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The objects of the Institution of Naval Architects—which was established to promote the Improvement of Ships, and of all that specially appertains to them—are comprised under three heads:—

First, the bringing together of those results of experience which so many shipbuilders, marine engineers, naval officers, yachtsmen, and others acquire, independently of each other, in various parts of the country, and which, though almost valueless when unconnected, doubtless tend much to improve our Navies when brought together in the printed Transactions of an Institution.

Secondly, the carrying out, by the collective agency of the Institution, of such experimental and other inquiries as may be deemed essential to the promotion of the science and art of shipbuilding, but are of too great magnitude for private persons to undertake individually.

Thirdly, the examination of new inventions, and the investigation of those professional questions which often arise, and were left undecided before the establishment of this Institution, because no public body to which professional reference could be made then existed.

BYE-LAWS AND REGULATIONS.

CONSTITUTION.

1. The Institution of Naval Architects shall consist of four classes, viz., Members, Associate Members, Associates, and Honorary Members.

2. *Members.*—The Class of Members shall comprise every person who on the 22nd March, 1899, was on the Register of Members, and every Naval Architect or Marine Engineer thereafter elected or transferred into the class of Members.

The Class of Associate Members shall comprise every Naval Architect and Marine Engineer who, under the provisions of Rules 43 and 44, may be elected into this class.

3. *Associates.*—The Class of Associates shall consist of persons who are qualified either by profession or occupation, or by scientific or other attainments, to discuss with Naval Architects the qualities of a ship, or the construction, manufacture, or arrangement of some part or parts of a ship or her equipment.

4. *Honorary Members.*—The Class of Honorary Members shall consist of persons upon whom the Council may see fit to confer an honorary distinction.

ELECTION AND DUTIES OF OFFICERS.

5. The Officers of the Institution shall consist of a President, Past Presidents, Vice-Presidents, Members of Council, Associate Members of Council (not exceeding in number one-third the number of Members of Council) a Treasurer, two Auditors of Accounts, and a Secretary or Secretaries.

6. A General Meeting of the Members, Associate Members, and Associates of the Institution shall be held annually before Easter in each year; and at this Annual General Meeting the Members of Council, Associate Members of Council, Treasurer, and Auditors for the ensuing year shall be elected.

7. At the Annual General Meeting Members and Associate Members only shall vote in the Election of Members of Council, and both Associates, Members, and Associate Members in the election of Associate Members of Council, the Treasurer, and the Auditors.

8. *President.*—Both Members and Associates of the Institution shall be eligible for election as President. The President shall preside over all meetings of the Institution, and of Officers of the Institution, at which he is present, and shall regulate and keep order in the proceedings.

9. *Vice-Presidents.*—Both Members and Associates of the Institution shall be eligible for election as Vice-Presidents. In the absence of the President, one of the Past Presidents or one of the Vice-Presidents shall preside at the General Meetings of the Institution, and shall regulate and keep order in the proceedings.

10. In case of the absence of the President, Past Presidents, and of all the Vice-Presidents, the Meeting may elect any Member of Council or Associate Member of Council, and in case of their absence any Member present to preside.

11. The Chairman at any Meeting of the Council of the Institution, when the votes of the Meeting, including his own, are equally divided, shall be entitled to give a casting vote.

12. Persons holding the office of Vice-President shall at all times be entitled to sit and vote with the Council.

13. *Past Presidents and Vice-Presidents.*—All Members who have held the posts of President or Vice-President shall, while their connection with the Institution as Members lasts, be entitled to sit and vote with the Members of Council.

14. *Members of Council.*—Members only shall be eligible for election as Members of Council at the Annual General Meeting.

15. *Associate Members of Council.*—Associates only shall be eligible for election as Associate Members of Council at the Annual General Meeting.

16. The Direction and Management of the Institution shall be vested in the Council for the time being, the Associate Members voting with the Members of Council in all cases, except in the decision of questions directly affecting the forms of ships and the construction of their hulls.

17. The Council shall meet as often as the business of the Institution requires, and at every Meeting five Members of the Council shall form a *quorum.*

18. The Council may appoint Committees to report to them upon special subjects.

19. All questions shall be decided in the Council by vote; but at the desire, expressed in writing, of any four Members or Associate Members present, the determination of any subject shall be postponed to the succeeding meeting of the Council.

20. An annual statement of the funds of the Institution, and of the receipts and payments of the past year shall be made under the direction of the Council, and, after having been verified and signed by the Auditors shall be laid before the Annual General Meeting.

21. The Council shall draw up an Annual Report on the state of the Institution, which shall be read at the Annual General Meeting.

22. It shall be the duty of the Council to adopt every possible means of advancing the Institution, to provide for properly conducting its business in all cases of emergency, such as the death or resignation of Officers, and to arrange for the publication of the Papers read at the Meetings, or of such documents as may be calculated to advance the objects of the Institution.

23. *Treasurer.*—Only Bankers, or Members of Council, or persons who have been Members of Council and are still Members of the Institution, shall be eligible for election as Treasurer.

24. *Trustees.*—There shall be four Trustees, two of whom shall be the President and Treasurer of the Institution for the time being. The remaining two shall be appointed by, and hold office at the pleasure of the Council. In the names of these trustees, under the direction of the Council of the Institution, all securities shall be taken and investments made, the whole of such property being, notwithstanding, subject to the disposition of the Council, and the order of the Council in writing, signed by the Chairman of the Meeting and countersigned by the Secretary, shall be obligatory upon and full authority for the Trustees.

25. *Auditors.*—All Members, Associate Members, and Associates of the Institution shall be eligible for election as Auditors.

26. The Auditors shall have access at all reasonable times to the Accounts of the pecuniary transactions of the Institution; and they shall examine and sign the annual statement of the Accounts before it is submitted by the Council to the Annual General Meeting.

27. *Secretary.*—The Secretary or Secretaries shall be elected by the Council, and shall be removable at the will of the Council, after due notice given. The salary of the Secretary or Secretaries shall be fixed by the Council.

28. It shall be the duty of the Secretary, under the direction of the Council, to conduct the correspondence of the Institution; to attend all Meetings of the Institution and of the Council; to take Minutes of the proceedings of such Meetings; to read the Minutes of the preceding Meeting; to announce donations made to the Institution; to superintend the publication of such Papers as the Council may direct; to have charge of the library, museum, and offices of the Institution; and to direct the collection of subscriptions and the preparation of accounts. He shall also engage, and be responsible for, all persons employed under him, and generally conduct the ordinary business of the Institution.

29. In each year six Ordinary and two Associate Members of Council shall retire, unless before the date of drawing up the Balloting Lists for the election of the Council any Members of the Council shall have died or resigned, in which case only so many members shall retire as shall be necessary in order to make up the number to six Ordinary and two Associate Members of Council, subject always to the provisions of Rule 36. The Members who shall retire in each year shall be those who have served longest on the Council from the date of the last election, and in the event of there being several Members who have served an equal time on the Council, the order of retirement amongst these shall be alphabetical. The retiring Members shall be eligible for re-election.

30. In January of each year the Council shall meet and prepare Lists for the election of the Council for the ensuing year. These Lists shall be as follow, namely :—

1st. A List of the names of the President, Vice-Presidents, and Treasurer for the ensuing year to be submitted at the Annual General Meeting, for their election in a body.

2nd. A list of candidates to fill any vacancies in the list of Vice-Presidents that it may be intended to fill up from among the professional Members of the Institution.

3rd. Lists for the election of the Ordinary Members and Associate Members of Council.

31. No addition shall be made to the list of Vice-Presidents until, by death or resignation, their number shall have been reduced to below twenty-four, after which their numbers shall be raised to and preserved at twenty-four. The vacancies are to be filled up in such a manner that not less than one-half nor more than two-thirds of the total list of Vice-Presidents shall be professional Members of the Institution. Provided always that the Council shall be at liberty, should special circumstances arise before the numbers shall have been reduced below twenty-four, to provide for the election of one Member and one Associate of the Institution as Vice-Presidents.

32. When any vacancy occurs in the list of Vice-Presidents which it is intended to fill by the election of a professional Member of the Institution, the election shall be by voting papers issued to all Members and Associate Members of the Institution. The candidates to fill the vacancy shall be selected by the Council in the month of January from among the existing or past Members of Council. The number of candidates shall not be less than two for each vacancy, but shall not otherwise be limited. The voting papers shall be issued to the Members and Associate Members at the same time as the voting papers for the election of Members of Council, and shall be subject to the same regulations and scrutiny as these latter, as provided for by Rules 37, 38, 39, 40.

33. After having been once elected by voting papers, the Vice-Presidents will be subject to re-election every year in a body at the Annual Meetings.

34. When any vacancy occurs in the list of Vice-Presidents which it is intended to fill by the election of an Associate of the Institution, the nomination to fill the vacancy shall be made by the Council, and the candidate nominated shall be included in the list of Vice-Presidents submitted at the Annual Meetings for election in a body.

35. No addition shall be made to the total number of Ordinary Members of the Council until, by death of resignation, their numbers shall have been reduced below twenty-four, after which their numbers shall be raised to and preserved at twenty-four. And no addition shall be made to the Associate Members of Council until, by death or resignation, their numbers shall have been reduced below eight, after which their numbers shall be raised to and preserved at eight, always exclusive of the President, Vice-Presidents, and Treasurer.

36. At the date of issuing the Syllabus of the Annual General Meetings in each year, the Lists proposed by the Council for the election of Members to fill the vacancies in the Ordinary Council for the ensuing year shall be printed, and sent to all Members and Associate Members to serve as Balloting Lists. These Lists shall contain, first, the names of the retiring Ordinary Members of Council at the time of the preparation ot the Balloting List, together with as many new names of Members of the Institution as shall be needed to bring the number up to twice the number of vacancies, and the whole of these names shall be printed in alphabetical order. Secondly, the names of the retiring Associate Members of Council at the time of the preparation of the Balloting List, together with as many new names of Associates of the Institution as shall be needed to bring the number up to twice the number of vacancies, and these names also shall be printed in

alphabetical order. From these Lists the vacancies in the Council shall be filled up. Every Member and Associate Member shall be at liberty to vote for as many names on each of the Lists as there are vacancies to be filled, but not for more.

37. A similar Balloting List (in which, however, the names of the Ordinary Members of Council proposed for election shall not be included) shall be printed and sent to all Associates of the Institution, to serve as a Balloting List for Associates, from which the voting for Associate Members of Council shall be taken. Every Associate shall be at liberty to vote for as many names on that List as there are vacancies to be filled, but not for more.

38. The Balloting Lists may be sent by post or otherwise to the Secretary, so as to reach him before the day and hour named for the Annual General Meeting, or they may be personally presented by the Members, Associate Members, and Associates at the opening of the Annual General Meeting.

39. At the opening of the Annual General Meeting the order of business shall be :—

(1) To read and consider the Reports of the Council and Treasurer.

(2) To read the List of Officers and Nomination for Council for the ensuing year, proposed by the Council.

(3) The Chairman shall next put to the Meeting the List containing the names of the President, Vice-Presidents, and Treasurer for election for the ensuing year.

(4) The Chairman shall then nominate two Scrutineers (of whom one only shall be a Member of the existing or proposed Council), and shall hand to them the Ballot Boxes containing the Voting Papers for the Ordinary Members of Council and Associate Members of Council ; and

(5) The Scrutineers shall receive all Ballot Papers which may have reached the Secretary, and all others which may be presented by Members, Associate Members, or Associates at the Meeting. The Scrutineers shall then retire and verify the Lists, and count the votes ; and shall, not later than the following day, report to the Chairman the names which have obtained the greatest number of votes, subject to the conditions of the Ballot. The Chairman shall then read the List presented by the Scrutineers, and shall declare the gentlemen named in the List to be duly elected, provided always that the List does not contain more names than there are vacancies to be filled. If, in consequence of two or more of the candidates receiving an equal number of votes, the List shall contain more names than there are vacancies, the Council shall, at their next meeting, decide which of these candidates shall be elected.

(6) After the Ballot shall have been taken, and the Scrutineers have retired, the Meeting will proceed to the other business before it.

40. The new Council and Officers shall take office immediately after the close of the Annual General Meeting.

41. In the event of any vacancy occurring in the offices of either President or Treasurer after the date of the Annual Election in any year, the Council shall have power to elect a new President or Treasurer as the case may be, who shall hold office till the conclusion of the next Annual General (Spring) Meeting of the Institution.

DESIGNATION OF MEMBERS AND ASSOCIATES.

42. Any Member, Associate Member, Associate, or Honorary Member, having occasion to designate himself as belonging to the Institution, shall state the class to which he belongs according to the following abbreviated forms, viz., M.I.N.A.; A.M.I.N.A.; Associate I.N.A.; Hon. Mem. I.N.A.

ELECTION OF MEMBERS, ASSOCIATE MEMBERS, AND ASSOCIATES.

43. *Admission of Members.*—Every Candidate for admission or for transference into the Class of Member shall be more than twenty-five years of age. Every Candidate for admission or for transfer into the Class of Associate Member shall be more than twenty-two years of age. Every Candidate in either of these classes shall comply with the following Regulations:—

He shall submit to the Council a statement showing that he has—

(*a*) Served an apprenticeship, or pupilage, in shipbuilding or marine engineering for at least five years, or

(*b*) Undergone a mixed training of at least four years, partly in shipbuilding or marine engine works, and partly in a School of Naval Architecture or Marine Engineering recognised by the Council of the Institution.

He shall further show that, after completing his training, as set forth under headings (*a*) or (*b*), he has been professionally employed in some public or private shipbuilding yard, or marine engine works, or registration society, or as professional superintendent of a steamship company, and shall give full particulars of the nature and duration of such employment.

In the event of the Candidate not having complied with any of the conditions under headings (*a*) or (*b*), he shall show—

(*c*) That he has been regularly educated by apprenticeship, or pupilage, or in a technical school, as a civil or mechanical engineer, and that he has been subsequently *professionally* employed for at least seven years in some public or private shipbuilding yard or marine engine works.

This statement shall be signed by at least three Members of the Institution, whose signatures shall certify their personal knowledge of the Candidate, and approval of his statement.

44. These preliminary conditions being satisfied, the Council shall then decide whether the education, the practical experience, and the professional attainments of the Candidate are such as entitle him to be recommended by the Council for election to the Institution, and, if so, whether as a Member, or as an Associate Member. The proposal of the Candidate for admission to the class decided on by the Council shall be submitted to the Members of the Institution (who shall have access to the applicant's statement), at an Ordinary Meeting of the Institution, for them to vote upon, the voting to be by ballot, should a ballot be demanded.

45. *Admission of Associates.*—Candidates for Associateship shall submit to the Council a proposal for their admission, setting forth therein a statement of their claims to be admitted as Associates. This proposal shall be signed by at least one Member, Associate Member, or Associate of the Institution. Their proposal, if approved by the Council, shall be submitted by them at an Ordinary Meeting of the Institution, for the Members, Associate Members, and Associates jointly to vote upon, the voting to be by ballot, should a ballot be demanded.

46. The proportion of votes for deciding the election of Members, Associate Members, and Associates shall be at least four-fifths of the numbers recorded.

SUBSCRIPTIONS.

47. Each Member, Associate Member, and Associate shall pay an Entrance Fee of two guineas, and an Annual Subscription of two guineas in advance; the first Subscription being payable on his election, and all future ones on the 1st day of January of each year. Any Member or Associate withdrawing from the Institution after that date is still liable for the amount of Subscription due on that day.

48. Any Member, Associate Member, or Associate may compound for his Annual Subscription, for life, by a single payment of not less than thirty guineas.

49. No person's name shall be entered on the Roll as Member, Associate Member, or Associate of the Institution nor possess the privileges of Membership (except it be on the honorary list) until he shall have paid his first subscription or the life composition, and if the payment be delayed for more than twelve months from the date of his election, the same shall be void unless the Council otherwise direct.

50. The Secretary shall at the close of every year notify to all Members, Associate Members, and Associates whose subscription for that year shall not have been paid, that it will be his duty to report accordingly to the Council, and he shall at the same time furnish the person whose subscription is in arrear with copies of this and the two following Rules.

51. The Secretary shall before Easter in every year lay before the Council a list of all Members, Associate Members, and Associates whose subscriptions for the two previous years shall be still unpaid, and unless the Council shall otherwise direct, the names of those in arrear shall be expunged from the Roll of Members, Associate Members, and Associates, and shall not be replaced without re-election in due form. Provided always that the Council shall at any time within two years therefrom have power to dispense with such re-election, and to restore the name to the Roll upon payment of all subscriptions then due, and upon cause being shown to the satisfaction of the Council why such subscriptions were not previously paid.

52. Nothing herein contained shall prejudice the right of the Institution to the legal recovering of all arrears of subscriptions up to the date of striking the name off the Roll.

53. In case the Council shall be of opinion that any Member, who has been long distinguished in his professional career, from ill-health, advanced age, or other sufficient causes, should not be called upon to continue his annual subscription, they may remit it. Also they may remit any arrears which are due from an individual, or may accept a collection of books, or drawings, or models, or other such contribution as, in their opinion, under the circumstances of the case, may entitle the person to be enrolled as a Life Subscriber, or to enable him to resume his former rank in the Institution which may have been in abeyance from any particular causes. These cases must be considered and reported upon by a Sub-Committee named for the purpose.

54. In case the expulsion of any individual shall be judged expedient by ten or more Members, and they think fit to draw up and sign a proposal requiring such expulsion, the same being delivered to the Secretary shall be by him laid before the Council. If the Council, after due inquiry, do not find reason to concur in the proposal, no entry thereof shall be made in any Minutes, nor shall any public discussion thereon be permitted; but if the Council do find good reason for the proposed expulsion they shall direct the Secretary to address a letter to the person proposed to be expelled, advising him to withdraw from the Institution. If that advice be followed, no entry on the Minutes nor any public discussion on the subject

shall be permitted; but if that advice be not followed, nor a satisfactory explanation given, the Council shall call a Special General Meeting of Members, Associate Members, and Associates, for the purpose of deciding on the question of expulsion; and if two-thirds of the persons present at such Special General Meeting, providing the number so present be not less than thirty, vote that such individual be expelled, the Chairman of that Meeting shall declare such expulsion accordingly, and the Secretary shall communicate the same to he individual.

MEETINGS.

55. Meetings for the Reading of Papers shall be held as frequently, and at such times, as the Council may determine.

TRANSACTIONS.

56. The *Transactions* of the Institution, including the Papers read at the Ordinary Meetings, and Reports of the Discussions by which they are followed, shall be edited by the Secretary, and printed under the direction of the Council.

57. A copy of each Volume of *Transactions* shall be sent free to every Member, Associate Member, and Associate.

58. The Secretary, under the direction of the Council, may dispose of the surplus stock of *Transactions* which have been published more than three years, at a price of not less than One Guinea a volume, provided a sufficient number remain on hand to supply the probable demand of New Members and Associates to complete their sets by the purchase of the back Volumes.

CHANGE OF ADDRESS.

59. Members, Associate Members, and Associates are particularly requested to communicate to the Secretary any change of address.

PROCEEDINGS IN LONDON.

SPRING MEETINGS OF THE FORTY-THIRD SESSION

OF THE

INSTITUTION OF NAVAL ARCHITECTS.

MARCH 19, 20, AND 21, 1902.

INTRODUCTORY PROCEEDINGS.

THE Spring Meetings of this, the Forty-third Session of the Institution of Naval Architects, were held on March 19, 20, and 21, 1902, in the Hall of the Society of Arts, John-street, Adelphi, W.C.

The opening meeting was presided over by the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President of the Institution, who commenced the proceedings by calling on the Secretary, Mr. R. W. Dana, to read the Report of the Council, which was as follows:—

ANNUAL REPORT OF THE COUNCIL, 1902.

The first duty of the Council is to announce that, in consequence of his accession to the Throne, His Majesty the King has been obliged to resign his honorary membership of the Institution; His Majesty has, however, graciously consented to become its Patron.

The Council are happy to report the continued prosperity of the Institution; the finances, as shown in the Treasurer's Report, are in a satisfactory condition, and the number* of new members elected during the past year is an additional proof, if any were needed, of the vitality of the Institution of Naval Architects.

The year that has elapsed has been marked by several interesting events in the history of the Institution; the Summer Meeting that was held in Glasgow last June, during the time of the Exhibition there, proved a most successful one, both on account of the cordiality of the reception extended to the members by the Corporation and the citizens of Glasgow, and also because of the importance of the questions raised and discussed at that meeting. Some of these, it is hoped, will lead to permanent benefits both to the country and to the profession.

In addition to the many objects of interest brought together by the Exhibition, all the leading shipbuilding yards and engineering works had very kindly arranged for Members to visit their works, either in groups or individually, during the time of the meeting. Members were welcomed at

* **Number of new Members, Associate Members, and Associates elected**	**121**
Number of deaths and resignations	**60**
Net gain in membership	**61**

a splendid reception, held on the first evening, by the Lord Provost and the Corporation of the City of Glasgow in their palatial Municipal Buildings, and the week closed with a most enjoyable day, spent at the invitation of the President and the Countess of Glasgow, at their charming seat, Kelburne, overlooking the most beautiful part of the Clyde.

A direct outcome of this Glasgow Meeting was the formation of a committee to inquire into the question of mercantile auxiliaries, dealt with in Lord Brassey's paper. The Council have the pleasure of announcing that, at their request, the President has laid the matter before the First Lord of the Admiralty, who has kindly promised to give full consideration to the views of the Institution.

A second committee has been formed to consider the possibility of establishing an experimental tank for the use of all shipbuilders throughout the kingdom, where models of proposed designs could be tested for resistance. The members of this committee are preparing a scheme for putting into effect the resolution unanimously passed at Glasgow in favour of establishing such a tank at an early date.

The Institution also took part in an International Engineering Congress, consisting of nine different sections, which was held in Glasgow in September; Section IV. (Naval Architecture and Marine Engineering) was under the auspices of the Institution, and the chair was accordingly taken by the President. Several papers of interest were read, and those Members who attended the Congress had the advantage of renewing acquaintance with all the features of interest that had been brought to their notice at the time of the Summer Meeting in June.

The Martell Scholarship, the regulations for which were set forth in last year's Report, has now been thrown open to competition, and the Council are glad to say that a good number of candidates have applied to compete. The scholarship will be awarded on the result of the evening examinations of the Board of Education, which will shortly be held.

The Council have had much pleasure in awarding the Gold Medal of the Institution to Professor W. E. Dalby, M.A., B.Sc., for his very valuable paper on "The Balancing of the Reciprocating Parts of Engines"; and a Premium of Books to Captain G. W. Hovgaard, for his paper on "The Motion of Submarine Boats in the Vertical Plane."

An invitation from the Schiffbautechnische Gesellschaft to the members of the Institution to take part in the Summer Meeting of the German Naval Architects at Düsseldorf next June has been received by the Council and accepted on behalf of the members, to whom a detailed programme of the meeting will shortly be sent.

The Council recall with much regret the deaths, which have occurred since last report, of Sir Raylton Dixon and Sir James Laing, both Members of Council and long connected with the Institution; of Admiral of the Fleet Sir John Edmund Commerell, G.C.B., V.C., a distinguished Associate; of Captain L. von Sztranyavszky, late Naval Attaché to the Austro-Hungarian Embassy; and of Messrs. H. Sandison and R. Barnard, whose melancholy deaths in the *Cobra* disaster last September are still fresh in the recollection of members.

Lastly, the Council have much pleasure in stating that the Testimonial to their late Secretary, Mr. George Holmes, has met with a very cordial response from the members of the Institution, and that the silver tazza designed and executed by Mr. Gilbert Marks at the Council's request, will be presented to Mr. Holmes, together with an illuminated address and a cheque for 1,000 guineas, at the annual general meeting.

TEXT OF THE ILLUMINATED ADDRESS PRESENTED TO MR. GEORGE HOLMES.

WE, the President, Vice-Presidents and Officers, Members, Associate Members, and Associates of the Institution of Naval Architects, desire to record on this Testimonial our high appreciation of your invaluable services as Secretary of the Institution during a period of twenty-three years, and in presenting you with the accompanying piece of plate as a token of our esteem and gratitude, we desire to express our hearty good wishes for a happy future for yourself and Mrs. Holmes.

We recognise that the present prosperity of the Institution is due, in no small degree, to the ability you have displayed in the discharge of your duties as Secretary, and we recall with much pleasure the various meetings, both at home and abroad, which you organised with such conspicuous success, and which have tended so greatly to promote harmony among our members and cordial relations with our foreign colleagues. May the success which crowned your labours at this Institution follow you to your present sphere of action, and may the friends whom you will make in your new surroundings prove as numerous and true as those you have left behind!

Signed, on behalf of the Institution of Naval Architects,

GLASGOW, *President.*
NATHANIEL BARNABY, *Vice-President.*
HENRY MORGAN, *Vice-President.*
R. W. DANA, *Secretary.*

Institution of Naval Architects.

Dr. *Statement of Receipts and Payments for the Year* 1901. Cr.

Receipts.	£	s.	d.	£	s.	d.
To Balance at Banker's, Dec. 31, 1900	457	9	0			
„ Balance in Secretary's hands ...	72	1	7			
				529	10	7
„ Annual Subscriptions				3,083	3	9
„ Life Subscriptions...				94	10	0
„ Admiralty Grant				250	0	0
„ Sale of Volumes				215	12	4
„ Sale of Silver Badges				23	4	9
				£4,196	1	5

Payments.	£	s.	d.	£	s.	d.
By Rent	220	0	0			
„ Housekeeper and Cleaning ...	37	14	0			
„ Advertising	14	5	2			
				271	19	2
„ Despatch of Volumes	75	12	8½			
„ Silver Badges	21	16	6			
„ Reporting	120	0	0			
				217	9	2½
„ Salaries	1,220	13	2			
„ Postages and Telegrams... ...	73	9	8			
„ Petty Disbursements	76	8	2½			
„ Banker's Charges	2	13	4			
				1,373	4	4½
„ Miscellaneous	130	3	9			
„ Balance of Preparation General Index	50	0	0			
„ Printing Volume XLIII., Papers, Circulars, &c....	1,218	17	6			
„ Translation	2	9	0			
				1,401	10	3
„ Travelling	111	9	1			
„ Expenses of Spring and Summer Meetings	79	3	11			
„ Gold Medal	12	10	0			
				203	3	0
„ Insurance...	2	10	0			
„ Investments	461	19	0			
„ Stationery...	19	12	0			
				484	1	0
„ Balance at Banker's Dec. 31, 1901	240	7	8			
„ Balance in Secretary's hands to meet Current Expenses ...	4	6	9			
				244	14	5
				£4,196	1	5

LIBRARY FUND.

Receipts.	£	s.	d.
To Balance at Banker's, December 31, 1900 ...	67	8	2
„ Entrance Fees	256	4	0
„ Interest on Investments	241	2	8
	£564	14	10

Payments.	£	s.	d.
By Library Expenses	23	15	9
„ Furniture and Repairs	24	6	3
„ Investments	300	0	0
„ Balance at Banker's, December 31, 1901 ...	216	12	10
	£564	14	10

H. MORGAN, Honorary Treasurer.
R. W. DANA, Secretary.

We have examined the above-written entries with the books and vouchers, and find them correct.

BALL, BAKER, DEED, CORNISH & Co.,
Chartered Accountants,
1, *Gresham Buildings, E.C.*

The following is a List of Donations to the Library:—

"Annual Report of the Royal National Lifeboat Institution," 1902. *Presented by the Royal National Lifeboat Institution.*

"Balancing of Engines." *Presented by the Author,* Professor W. E. DALBY, M.A., B.Sc.

"Bureau Veritas International Register of Shipping," 1902. *Presented by the Council of the Bureau Veritas International Register of Shipping.*

"Bulletin de la Société Scientifique et Industrielle de Marseille," 1902. *Presented by the Editor.*

"Bulletin de la Société d'Encouragement pour l'Industrie Nationale," 1902. *Presented by the Société d'Encouragement pour l'Industrie Nationale.*

"Calendar of the University College of South Wales and Monmouth," 1901-02. *Presented by the Registrar of the University College of South Wales and Monmouth.*

"Chaudières Marines." *Presented by the Author,* Mons. E. BERTIN.

"Congrès International d'Architecture et de Construction Navales. Procès-verbaux Sommaires." *Presented by the Ministère du Commerce, Paris.*

"Germanischer Lloyd: International Register for the Year 1902." *Presented by the Committee of the Germanischer Lloyd.*

"International Engineering Congress, Glasgow," 1901. Sections I. (Railways), II. (Waterways and Maritime Works), and VIII. (Gas).

"Jahrbuch der Schiffbautechnischen Gesellschaft," 1901. *Presented by the Schiffbautechnische Gesellschaft.*

"Journal of the Franklin Institute," 1901. *Presented by the Franklin Institute.*

"Journal of the Iron and Steel Institute," 1901. *Presented by the Iron and Steel Institute.*

"Journal of the Royal United Service Institution," 1901. *Presented by the Royal United Service Institution.*

"Journal of the Society of Arts," 1901. *Presented by the Society of Arts.*

"Journal of the United States Artillery," 1901. *Presented by the Editor of the Journal.*

"Journal of the Western Society of Engineers," 1901. *Presented by the Western Society of Engineers.*

"Lloyd's Register of British and Foreign Shipping," 1902-03. *Presented by the Committee of Lloyd's Register.*

"Lloyd's Register of Yachts," 1902. *Presented by the Committee of Lloyd's Register.*

"Local Industries of Glasgow and West of Scotland." *Presented by the British Association.*

"Mémoires et Compte-Rendu des Travaux de la Société des Ingénieurs Civils de France," 1901. *Presented by the Société des Ingénieurs Civils de France.*

"Mémorial du Génie-Maritime. Troisième série." *Presented by the Ministère de la Marine, Paris.*

"Minutes of the Proceedings of the Institution of Civil Engineers," Vols. CXLIV., CXLV., CXLVI., and CXLVII. *Presented by the Institution of Civil Engineers.*

"Philosophical Transactions of the Royal Society of London," Vols. CXCII.—CXCVI., "A" Series. *Presented by the Royal Society of London.*

"Photographs of Members of the Institution." *Presented by the Photographers,* Messrs. MAULL & FOX, Piccadilly.

"Proceedings of the Institution of Mechanical Engineers," 1901. *Presented by the Institution of Mechanical Engineers.*

"Proceedings of the London Mathematical Society," 1901. *Presented by the London Mathematical Society.*

"Proceedings of the Royal Society of London," Vols. LXVIII. and LXIX., 1901-2.

"Proceedings of the Royal Society of New South Wales," Vol. XXXIV., 1900. *Presented by the Royal Society of New South Wales.*

"Proceedings of the United States Naval Institute," 1901. *Presented by the U.S. Naval Institute.*

"Report of Tests of Metals and other Materials made at Watertown Arsenal," for the year ending June 30, 1900. *Presented by the Chief of Ordnance, U.S.A.*

"Revista Marittima," 1901. *Presented by the Ministero della Marina.*

"Royal Society's Year Book," 1902. *Presented by the Royal Society.*

"Strength of Boiler Shells," 1875; "Shafting of Screw Steamers," 1887; "Unusual Corrosion"; 3 *pamphlets. Presented by the Author,* HECTOR MACCOLL, Esq.

"Technology Quarterly and Proceedings of the Society of Arts," 1901. *Presented by the Massachusetts Institute of Technology.*

"The Naval Annual," 1902. By the Right Hon. Lord BRASSEY, K.C.B., and others. *Presented by the Editor, the Hon.* T. A. BRASSEY.

"The Scientific Proceedings of the Royal Society of Dublin," 1901. *Presented by the Royal Society of Dublin.*

"Transactions of the American Institute of Mining Engineers," Vol. XXX. *Presented by the American Institute of Mining Engineers.*

"Transactions of the American Society of Mechanical Engineers," Vol. XXII., 1901. *Presented by the Council of the American Society of Mechanical Engineers.*

"Transactions of the Institution of Engineers and Shipbuilders in Scotland," Vol. XLIV., 1899-1900. *Presented by the Institution of Engineers and Shipbuilders in Scotland.*

"Transactions of the Institution of Junior Engineers," Vol. X., 1900. *Presented by the Institution of Junior Engineers.*

"Transactions of the Institute of Marine Engineers," 1900-1901. *Presented by the Institute of Marine Engineers.*

"Transactions of the Liverpool Engineering Society," 1901. *Presented by the Liverpool Engineering Society.*

"Transactions of the North-East Coast Institution of Engineers and Shipbuilders," Vol. XVII., 1900-1901. *Presented by the North-East Coast Institution.*

"Transactions of the North of England Institute of Mining and Mechanical Engineers," 1901. *Presented by the North of England Institute of Mining and Mechanical Engineers.*

"Transactions of the Society of Engineers," 1901. *Presented by the Society of Engineers.*

"Transactions of the Society of Naval Architects and Marine Engineers," Vol. IX., 1901. *Presented by the Society of Naval Architects and Marine Engineers.*

"Water Tube Boilers." *Presented by the Author,* LESLIE J. ROBERTSON, Esq.

Newspapers and Magazines, presented by the respective Proprietors :—

"Annalen für Gewerbe und Bauwesen," 1901.
"Arms and Explosives," 1901.
"Army and Navy Gazette," 1901.
"Automotor and Horseless Vehicle Journal," 1901.
"Cassier's Magazine," 1901.
"Commerce," 1901.
"Engineer," 1901.
"Engineering," 1901.
"Engineering Times," 1901.
"Engineers' Gazette," 1901.
"English Mechanic," 1901.
"Field," for 1901.
"Fielden's Magazine," 1901.
"Invention," 1901.
"Iron and Coal Trades Review," 1901.
"Journal of the Imperial Institute," 1901.
"Le Moniteur de la Flotte," 1901.
"Machinery Market," 1901.
"Marine Engineer," 1901.
"Marine Engineering," 1901.
"Marine Review," 1901.
"Mariner," 1901.
"Mechanical Engineer," 1901.
"Practical Engineer," 1901.
"Railway Engineer," 1901.
"Saturday Review," 1901.
"Science Abstracts," 1901.
"Shipping World," 1901.
"Zeitschrift des Vereins Deutscher Ingenieure," 1901

The following gentlemen, having been duly recommended by the Council, were unanimously elected Members of this Institution :—

ANDERSON, J. HALL, Manager to Messrs. Hall, Russell & Co., *Aberdeen.*
BAILLIE, ROBERT, Superintending Engineer to the Stirling Boiler Co., Ltd., *Glasgow.*
BELL, JAMES, Chief Assistant Shipyard Manager to Messrs. W. Beardmore & Co., Govan Yard, *Glasgow.*
BIGGAR, C. J., Naval Architect, *Swanwick, near Southampton.*
BLAIR, A., Superintending Engineer to the Anglo-American Oil Co., *Billiter Buildings, London, E.C.*
BOOBNOFF, IVAN G., Assistant to the Director of the Imperial Russian Naval Tank, *St. Petersburg.*
BOULTON, THOMAS, Naval Architect, *Newcastle-on-Tyne.*
BRYSON, JOHN J., Manager to the Bergens Mekaniske Vœrksted, *Bergen, Norway.*
DITCHBURN, ROBERT, Manager of the Bombay Steam Navigation Co., *Bombay, India.*
DIXON, WAYNMAN, Managing Director to Messrs. Sir Raylton Dixon & Co., *Middlesboro'.*
FERNIE, JAMES, Consulting Engineer and Naval Architect, *Liverpool.*
HANSCOM, C. R., President, Eastern Shipbuilding Co., *New London, Conn., U.S.A.*
HIDE, W. SEYMOUR, Superintending Engineer to Messrs. Thos. Wilson, Sons, & Co., *Hull.*
HUTCHINGS, T. C., Manager to the Hong Kong and Whampoa Co., *Hong Kong, China.*
KERFOOT, JAMES, Manager of the Technical Department of the Antwerp Engineering Co., Ltd., *Antwerp.*
NICHOLSON, C. ERNEST, of Messrs. Camper Nicholson, Ltd., *Gosport.*
O'NEILL, JAMES J., Chief Draughtsman to the Fairfield Shipbuilding and Engineering Co., *Govan.*
PHILIP, G. NOWELL, General Manager of the Sandquay Engineering Works, *Dartmouth.*
PRUNERI, G., of the Construction Department, Ministry of Marine, *Rome.*
REED, JOSEPH, Marine Surveyor, *Newcastle-on-Tyne.*
SCHOFIELD, CHARLES, Superintendent of the Eastern Shipbuilding Co., *New London, Conn., U.S.A.*
SHARP, WILLIAM E., Naval Architect and Consulting Engineer, *Bangkok, Siam.*
SIM, JOHN A., Naval Architect to Messrs. Sir James Laing & Sons, Ltd., *Sunderland.*
SKENTELBERY, CHARLES, Manager to Messrs. Jacobs & Barringer, *Boston, U.S.A.*
WILKINSON, THOMAS, Chief Draughtsman to Messrs. C. S. Swan & Hunter, Ltd., *Wallsend-on-Tyne.*
WILLIAMSON, A. SMART, Surveyor to Lloyd's Register of Shipping, *Yokohama.*
WINTERBURN, W. G., General Manager to Messrs. George Fenwick & Co., Ltd., *Hong Kong.*

The following gentlemen were elected Associate Members :—

ALLEN, H. PINNY, Naval Architect to Messrs. S. C. Farnham, Boyd & Co., *Shanghai, China.*
ANDERSON, GEORGE, Assistant to Mr. A. R. Brown, Consulting and Marine Engineer, &c., *Lime-street, E.C.*
BROWN, J. DOWIE, Joint Manager, The Rosebank Iron Works, *Edinburgh.*
DARBY, J. T., Naval Architect, *Bootle, Liverpool.*
HARTNOLL, A. E. N., Overseer to Messrs. Cox & King, Naval Architects, *Pall Mall.*
KEY, HENRY, Engineer in Charge of Workshops and Slipway at *Akassa, Southern Nigeria Colony.*
KIMURA, N., Draughtsman at the Fairfield Shipbuilding and Engineering Co., *Govan.*
KOSSURA, P. S., Assistant Manager, Nevsky Shipbuilding and Mechanical Works, *St. Petersburg.*
LEGARDA, V. L., Naval Architect at the Union Iron Works, *San Francisco, U.S.A.*
PEARCE, F. T., Yacht Draughtsman, Westmacott, Stewart & Co., *Isle of Wight.*
REIACH, H. L., General Manager to Messrs. J. G. Fay & Co., Ltd., *Southampton.*
WILSON, A. HALL, B.Sc., Managing Director, Messrs. Hall, Russell & Co., *Aberdeen.*

The following gentlemen were elected Associates:—

Belam, H.	Duncan, James.	Schwarz, Chevalier de, Comnder.
Blake, W. M.	Ewart, A. E., R.N.	Scribanti, A., Captain.
Brodie, G. W.	Hansen, H. S.	Segrave, T. G.
Brown, Alfred.	Henderson, W. H., Rear-Adm.	Seton, Bruce G., Captain.
Byers, W. L.	Irizar, J., Commander.	Stewart, T. Cuthbert.
Clowes, W. Laird.	MacBrayne, L.	Taylor, R. S., Captain.
Croix, P. de Ste.	Magill, H.	Tegelberg, G. L
Daw, H.	Petrie, David.	Tomikawa, N
Deane, R.	Roby, A. W.	Welin, Axel.

Since the issue of Volume XLIII., the Institution has sustained the loss by death of the following gentlemen:—

Vice-President.

Benjamin Martell.

Member of Council.

Sir James Laing.

Members.

Allfrey, E.	Cooke, R.	Sandison, M.
Barrie, W.	Guillaume, G. C. A.	Spence, J. A.
Barnard, R.	Holst, F. E.	Stainer, G. H., C.B.
Clark, G.	Laing, Arthur.	

Associates.

Cooper, J. H., Colonel	Stewart, Andrew.	Watson, W.
Smith, Eustace.	Sztranyavszky, L. Von, Capt.	Wills, W.

The Secretary next read the following list of names nominated by the Council to fill up the vacancies for the ordinary and Associate Members of Council for the ensuing year:—

Retiring Members of Council.

Adamson, A.	Barnaby, Sydney W.	Deadman, H. E.

New Candidates nominated for Members of Council.

Dunlop, J. G.	Soper, Thos.	Thompson, Robert.
Seaton, A. E.	Swan, H. F.	West, H. H.
Spence, H. G.	Thearle, S. J. P.	Withy, H.

Retiring Associate Members of Council.

Lewes, Professor V. B., F.R.S., F.I.C.	Riley, James.

New Candidates nominated for Associate Members of Council.

FitzGerald, C. C. P., Vice-Admiral.	Digby Morant, Admiral Sir G., K.C.B.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.) next put to the meeting the following list containing the names of the President, Vice-Presidents, and Treasurer for the ensuing year, which were unanimously adopted:—

President.

THE RIGHT HON. THE EARL OF GLASGOW, G.C.M.G.

Past Presidents.

THE RIGHT HON. THE EARL OF RAVENSWORTH.

THE RIGHT HON. LORD BRASSEY, K.C.B., D.C.L.

THE RIGHT HON. THE EARL OF HOPETOUN, P.C., K.T., G.C.V.O., G.C.M.G.

Vice-Presidents.

H.R.H. GEORGE, PRINCE OF WALES, K.G., K.T., K.P., G.C.V.O., G.C.M.G., Rear-Admiral.
THE RIGHT HON THE EARL OF NORTHBROOK, G.C.S.I.
THE RIGHT HON. THE EARL OF RAVENSWORTH.
THE RIGHT HON. EARL SPENCER, K.G.
THE RIGHT HON. LORD GEORGE HAMILTON, M.P.
THE RIGHT HON. LORD JOHN HAY, G.C.B., Admiral of the Fleet.
THE RIGHT HON. SIR JOHN DALRYMPLE-HAY, Bart., K.C.B., D.C.L., F.R.S., Admiral.
SIR NATHANIEL BARNABY, K.C.B.
SIR FREDERICK BRAMWELL, Bart., D.C.L., F.R.S.
SIR JOHN DURSTON, K.C.B., R.N.
SIR JOHN GLOVER.
SIR ANDREW NOBLE, K.C.B., F.R.S.
SIR EDWARD J. REED, K.C.B., M.P., F.R.S.
SIR W. H. WHITE, K.C.B., LL.D., Sc.D., F.R.S.
F. K. BARNES, ESQ.
JAMES DUNN, ESQ.
F. ELGAR, ESQ., LL.D., F.R.S.
JOHN INGLIS, ESQ., LL.D.
BENJAMIN MARTELL, ESQ.
HENRY MORGAN, Esq.
GEORGE W. RENDEL, Esq.
J. I. THORNYCROFT, Esq., F.R.S.
PHILIP WATTS, Esq., F.R.S.
A. F. YARROW, Esq.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, the Council, in the execution of their duties, have found that certain alterations should be adopted in Rules 5, 9 and 10, relating to Past Presidents, and I will ask Mr. Morgan, the Treasurer, to move the adoption of the alterations which are recommended by the Council.

Mr. HENRY MORGAN (Vice-President and Treasurer): My Lord and Gentlemen, a few words in explanation of the necessity of the alterations will perhaps be desirable. For many years, in fact until the last ten years or so, we had no Past Presidents. We had a President of the Institution, a Vice-President, and other officers; the rules which were drawn up at the commencement consequently make no provision for the duties or the privileges of Past Presidents. We have now in reality three Past Presidents still living, namely, Lord Ravensworth, Lord Brassey, and Lord Hopetoun. It has always been the desire of the Council that these Past Presidents should exercise the same privileges as the Vice-Presidents, and in practice that has been the case, although the rules make no definite provision for that. It is therefore proposed to-day to make certain small alterations in the rules, as stated in the papers which are, I think, in the hands of many of the members. This will enable Past Presidents to sit with the Council, to preside over meetings in the absence of the President, and to vote in the same way as Vice-Presidents do. These alterations have been carefully prepared by the Council in order to carry out that object. They are as follows:—

Rule 5.—*After the words* "of a President," *insert the words* "Past Presidents."

Rule 9.—*After the words* "one of the," *insert the words* "Past Presidents or one of the."

Rule 10.—*After the words* "of the President," *insert the words* "Past Presidents."

Rule 13.—*For the words* "President and Vice-President," *substitute the words* "President or Vice-President."

The Rules will then read:—

5. The Officers of the Institution shall consist of a President, Past Presidents, Vice-Presidents, Members of Council, Associate Members of Council (not exceeding in number one-third the number of Members of Council), a Treasurer, two Auditors of Accounts, and a Secretary or Secretaries.

9. *Vice-Presidents.*—Both Members and Associates of the Institution shall be eligible for election as Vice-Presidents. In the absence of the President, one of the Past Presidents or one of the Vice-Presidents shall preside at the General Meetings of the Institution, and shall regulate and keep order in the proceedings.

10. In case of the absence of the President, Past Presidents, and of all the Vice-Presidents, the Meeting may elect any Member of Council or Associate Member of Council, and in case of their absence any member present to preside.

13. *Past Presidents and Vice-Presidents.*—All Members who have held the posts of President or Vice-President shall, while their connection with the Institution as Members lasts, be entitled to sit and vote with the Members of Council.

Sir NATHANIEL BARNABY: I beg to second that.

The resolution was put to the meeting and carried unanimously.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.) then proceeded to deliver the following Opening Address:—

In the Report you have just heard read, allusion has been made to several questions of interest that have, during the past year, come under the consideration of the Council, either as the result of some of the noteworthy subjects discussed at the Meetings, or in response to invitations from kindred Institutions to join in the investigation of questions affecting other interests besides those of our own members. And should our labours result, as we hope they will, in the strengthening of certain branches of our auxiliary naval forces, or in providing means by which shipbuilders may improve upon their designs or simplify their work, we shall feel at least that, during the year that has passed, our Institution has continued to fulfil those useful functions which its founders laid down as the guiding principles of its existence.

It may be well to go rather more fully here into some of these questions than was possible in the Report of Council. It will be in the recollection of those of you who were present at the Summer Meeting at Glasgow, last June, and who heard Lord Brassey's able paper on "Mercantile Auxiliaries," that so much importance was attached to the question that the following resolution, proposed by Sir John Dalrymple-Hay, was unanimously agreed to :—

"That as our supremacy depends on the efficiency of our Naval and Mercantile Marine, a committee of Admiralty officials, shipowners, and shipbuilders should be formed to discuss the best method of constructing a combined Naval and Mercantile Marine."

The matter afterwards came up before the Council, who drew up a statement on the subject, giving the reasons which appeared to them to make the matter one of urgency. These reasons were embodied in the following letter which was sent to the first Lord of the Admiralty :—

INSTITUTION OF NAVAL ARCHITECTS,
5, *Adelphi Terrace, London, W.C.*,
January 14, 1902.

To the Rt. Hon. the Earl of SELBORNE,
First Lord of the Admiralty.

MY LORD,—I have the honour to inform you that, as President of the Institution of Naval Architects, I have been deputed by the Council of that body to approach you upon a subject which, in our humble opinion, merits the careful consideration of His Majesty's Government: namely, the future relations between the Government and the owners of fast liners and other vessels which are likely to be of use in time of war.

This question appears to us to be one of paramount importance, and we humbly submit that it is most desirable that Government should institute an inquiry into the whole question, by means of a competent Committee composed of shipowners and shipbuilders, Admiralty officials, Naval officers, and members of the Legislature.

In making this proposal, we have been guided by the following considerations :—

(1) That unless there is some undertaking between the State and the owners of fast liners and other desirable vessels to bind their ships to State service, these ships and vessels may be found unexpectedly in the hands of an enemy at the outbreak of war.

(2) That the contract should be of such a nature as would effectually deter owners from breaking it.

(3) That, with reasonable inducement, shipowners would set themselves to make the ships and vessels suitable for war purposes.

(4) That, by these means, the State Navy may be considerably strengthened without adding to the expenditure on building and maintaining war cruisers as distinct from the heavier fighting ships.

It is not practically possible adequately to protect a vast commerce with regularly built vessels of war only, and all the leading maritime Powers have consequently found it necessary to create a reserve of cruisers. As is well known to you, My Lord, the State has already, to a certain extent, made an advance in this direction, and it is in order that in future the leading vessels of the Mercantile Marine may not merely be subsidised for employment in case of war, but actually constructed with this end in view, that we venture to bring this question before your notice.

I enclose herewith, for your perusal, a paper on "Mercantile Auxiliaries," by Lord Brassey (Past President of this Institution), which sets forth in detail the reasons I have outlined above. At the conclusion of the discussion which followed the reading of that paper, a resolution was proposed by Admiral Sir John Dalrymple-Hay and carried unanimously, to the effect that such a Committee as that above referred to should be appointed by Government.

I venture, therefore, to bring this subject before your notice, and I trust that you may consider it of sufficient importance to justify an independent inquiry into the subject.

I have the honour to remain, My Lord,
Your obedient Servant,
(Signed) GLASGOW.

The day after the despatch of this letter, I had the honour of an interview with the First Lord of the Admiralty, and I have quite recently received a letter from Lord Selborne, in which he states that the Admiralty have come to the conclusion to appoint a small Committee consisting of three members only, one an Admiralty official, another a Treasury official, and the third engaged in the shipbuilding interest. I have been invited to assist Lord Selborne in forming that Committee, and I think it will be very satisfactory to the Institution to know that its representations to the Admiralty have been fruitful of this very good result.

With regard to the proposed establishment of an experimental tank for the use of shipbuilders throughout the country, the Committee which the Council selected to consider this question, applied in the first instance to the Admiralty in order to see if it would be possible, before seeking the desired accommodation elsewhere, to extend the use of the existing tank at Haslar, so that it might be made available for private shipbuilders, as is the case at the Government tanks at Washington, Spezia, and St. Petersburg, on payment of suitable fees. To this application the Secretary of the Admiralty replied that he had laid the matter before his Board, and that their Lordships, while sympathising with the movement, found it impossible, owing to the large demands on the capacity and staff of the existing tank at Haslar, to further extend its use. Their Lordships, however, kindly granted permission to Mr. R. E. Froude, who has so long been in charge of the Haslar tank, to give the Committee all necessary information regarding cost of construction and equipment, &c., to enable them to form estimates for a similar tank elsewhere.

The Committee, therefore, having considered various proposed locations, have thought that the National Physical Laboratory at Bushy House, where a very suitable site can be found, would offer many important advantages; a position conveniently near the Metropolis, the proximity of an established power installation, and a highly trained technical staff, and, above all, a board of management of the highest standing, entirely unconnected with any individual commercial enterprise, whose control would in itself guarantee the treatment in strictest confidence and impartiality of all questions submitted to them.

The Committee accordingly approached the General Board of the National Physical Laboratory with a view to ascertaining if the latter would favour the proposal of establishing and maintaining an experimental tank in the grounds of Bushy House, provided that a sufficient sum were placed at the disposal of the Board to build and equip such a tank, the responsibility of maintenance resting with the Laboratory Board, and the Council retaining a voice in the management of the tank to the satisfaction of the Institution.

The reply to this application is contained in the following Minute of the Executive Committee's meeting of the National Physical Laboratory :—

"The letter from the Committee of the Council of the Institution of Naval Architects as to the proposed Naval Tank having been read, it was agreed unanimously to instruct the Director to convey to the Committee of the Institution of Naval Architects the thanks of the Executive Committee for the proposed gift, and their hearty approval of the suggestion that the tank should be established at Bushy, and to state that, if financial considerations allow the scheme to be carried into effect, the Committee are prepared cordially to recommend its acceptance to the Council of the Royal Society."

It remains now for those members of the Institution who are interested in the matter to give effect to their former resolution, by voluntarily guaranteeing the necessary financial support, and the Council hope in due course to submit a proposal for your consideration.

Turning to the broader aspect of shipbuilding, the returns of mercantile tonnage launched during the year continue to show a steady growth over past years, and if, at the present moment, the output is somewhat in excess of the immediate demands, the great expansion that is taking place in the world's commerce promises to justify this increase at no distant date.

The 639 vessels of the Mercantile Marine launched during the year in the United Kingdom have aggregated 1,524,739 gross tons, or 82,368 tons more than last year, this being an increase of 5½ per cent. on the previous best year, while for the Royal Navy 41 vessels of a combined tonnage of 211,969 tons have been built, this being 10 per cent. more than the largest previous yearly output (1898).

The total tonnage under construction at the close of last year was 89,000 tons more (or 7 per cent. more) than the corresponding amount twelve months before.

But we must not lose sight of the fact that, great as has been the growth of British shipbuilding during recent years, other nations have entered the lists, and are straining every nerve in the great struggle for the markets of the world. In Germany, France, and the United States, we see our greatest rivals taking advantage of every advance in scientific research, every improvement in labour and time-saving devices, to outstrip us in the race; and in shipbuilding, their efforts seem likely to receive further extraneous assistance from the bounties which their Governments have either already established, or with which they contemplate fostering this branch of trade. Whether this assisting of an industry that apparently shows every sign of increased vitality will not, in the end, defeat its own objects, remains for the future to prove. The great strides that have been made in recent years in American shipyards are significant of the practical manner in which the subject has been taken up in that country, and British shipbuilders will do well to keep before them the need of constant thought and effort, and of improved methods unhampered by custom or conventionality.

The returns of German shipbuilding show that, exclusive of war vessels, 217,600 tons were built in 1901, which compares with 204,700 tons in 1900, or an increase of 6½ per cent; this includes such steamers as the *Kronprinz Wilhelm*, of 14,900 tons, and the *Blucher* and *Moltke*, of 12,372 tons each; while the 66,470 tons of warship construction launched in Germany during the past year is the highest return of all the foreign Powers for that year.

The United States output for 1901 is very large: 433,200 tons were launched, as compared with 333,500 in 1900, but this includes no less than 159,300 tons of steamers on the Great Lakes, and

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135,400 tons of wooden and sailing vessels, thus reducing the output of steel and iron Ocean tonnage proper to 138,000 tons. This latter figure compares with 71,423 tons for the previous year, the output of merchant (Ocean) tonnage having, therefore, nearly doubled, while the figures for warships are no less remarkable, 25,000 tons in 1900, and 48,000 tons in 1901.

In France, the returns also show a large increase upon last year's output, the figures being (exclusive of 54,861 tons of warships) 177,500 tons in 1901, compared with 116,850 tons in 1900, an increase of over 50 per cent. This includes the very large total of 124,600 tons of sailing vessels, a direct outcome of the heavy bounties on shipbuilding and ship-owning which the Government have seen fit to grant. This policy, which the French have so strongly adopted of late years, no doubt accounts for the large returns of tonnage built, but we must not forget that this is only one aspect of a much larger economic question, and one which affects a far wider range in a nation's welfare than merely its shipbuilding interests.

On this point, I may, perhaps, quote with advantage the following remarks from Sir John Glover's paper, "Tonnage Statistics of the Decade (1891-1900)":—

"France, I think, led the way in this system of bounties, and so far as appears, she has not gained much by it. Between 1890 and 1900, French imports and exports increased only £12,000,000 sterling, from £327,000,000 to £339,000,000. That is a poor result, seeing that the French Government paid during the ten years 1891-1900, in bounties, for construction and navigation, nearly £5,250,000 sterling, in addition to over £10,000,000 sterling in subventions for postal services. It is also significant that the total entries and clearances of French shipping at French ports in 1890 was 9,254,879 tons, and, in 1899, only 10,137,277 tons; and that the percentage of tonnage entered and cleared under the French flag in French ports fell from 31·9 in 1890, to 28·4 in 1899. The total tonnage of the French Mercantile Marine is given at 932,735 for 1889, and 957,755 for 1899. These cannot be regarded as encouraging facts for states which are contemplating the adoption of the bounty system against us."

In the field of marine engineering, the two reports of the Admiralty Water Tube Boiler Committee have caused the attention of engineers to be still engrossed by this much-discussed steam generator. Reference was made to the first report in my address delivered at the Spring Meeting of last year. The second report, issued last month, contains full details of some of the most elaborate trials ever made with vessels of war. Whatever may be the result of the investigation upon the type of boiler ultimately adopted for use in the Royal Navy, the information given in the report, in regard to the performance of the machinery, will be of great value to Marine Engineers of all classes. The report, which has been published in full by the technical press, will doubtless have been read by the majority of the Members of the Institution, and I hardly think it necessary for me to enter into the details of a subject so entirely technical, and upon which such diverse opinions appear to be still held. It may, however, be pointed out that, whatever may be the respective merits of the shell boiler and the water-tube boiler, or of different types of the latter, the navies of the world appear to have definitely decided to adopt the newer form of steam generator in one or more of its many forms.

In motive machinery afloat, the changes now being made are chiefly in details of construction. The triple-expansion engine is still being fitted in ships of the Royal Navy, but in the Mercantile Marine the fourth step in compounding is now frequently made. The newer ships for the American Navy are also to have quadruple-expansion engines.

One of the chief matters of interest among naval engineers at present is the steam turbine. It is so entirely different in principle and construction from the reciprocating cylinder and piston engine, that, whatever its merits may ultimately prove to be—and it undoubtedly is a practical marine motor—its appearance in the world of marine engineering must always remain an event of the greatest interest. During the past year, the marine steam turbine has been put to a commercial use in a very practical way. A passenger steamer, the *King Edward*, built by Messrs. Wm. Denny & Bros., of Dumbarton, and fitted with the new type of machinery by the Parsons Marine Steam Turbine Company, of Wallsend-on-Tyne, was put on the route between Fairlie and Campbeltown early in the summer, and was run on the regular service for the rest of the season, and the proprietors are constructing a larger and faster vessel of the same type.

Turning to another aspect of marine engineering afloat, one notices that the practical employment of liquid fuel for generating steam has also been a good deal canvassed of late. This is rather a commercial, or an economic problem, than an engineering one, as it is well known to those who have used liquid fuel on ocean voyages, that there is no difficulty whatever in burning it. The naval authorities of the different maritime Powers have given a good deal of attention to the question as applied to warships, and a large number of experiments have been made. Although the details of these have not been made public, it is generally reported that liquid fuel is looked upon as likely to prove a valuable auxiliary in case of war.

It is now some years since Sir Edward Reed drew attention to the advantages of liquid fuel. Whether those who are now endeavouring to establish this new departure in ocean steamships and warships—for new it would be in regard to its magnitude—will succeed in placing it on a working basis, time and experience will prove; but when one remembers the benefits that are promised—cleanliness, saving of labour, reduction in the number of stokers, economy of space, and reduced weight to carry—one must recognise that its success will be a great boon.

The equipment of shipyards is a problem of mechanical engineering which has occupied a good deal of attention of late. The development of electrical power appliances has opened up resources by which the shipyard manager may save in cost of construction and facilitate work at the same time. This applies not only to the driving of machinery, but also to the working of lifting appliances. The machine tools in a shipyard are widely spread, and often in exposed situations; so that there is considerable loss from condensation in the long lengths of pipe when they are steam-driven, and the engines are often kept turning for considerable periods when the machines are not in use, simply to prevent cylinders filling with water. Electricity naturally overcomes this difficulty. For lifting and transporting material, gantrys are erected to carry fast-running overhead electric travelling cranes; these are on the cantilever principle, and can serve two building slips.

Reverting to events closely connected with the welfare of this Institution, we cannot but note with satisfaction the evident desire of those connected with the Board of Admiralty to give due consideration to questions of importance which have arisen of late years. And the mention of the Admiralty recalls the change that has taken place in the Naval Construction Department. Sir William White's retirement, after a period of sixteen years at the head of that most important branch of the public service, marks an epoch in the history of our Naval Construction, the importance of which the members of the Institution will be the first to appreciate. None will be readier to pay a fitting tribute to the genius and the devoted public services of Sir William White, who has so long and so honourably been connected with the Institution. That his retirement should have been due to his health breaking down under the strain of his arduous duties can only increase our sympathies, and stimulate the hope that he may, with rest and change, regain strength and health; and that he may continue, as of old, to take part in the affairs of the Institution, and contribute, with his invaluable experience and judgment, to the interest of our meetings for many years to come.

Looking to the future, our thoughts turn to our esteemed Vice-President, Mr. Philip Watts, F.R.S., who succeeds Sir William White as Director of Naval Construction. Mr. Watts' connection with the Institution dates back almost as far as that of Sir William White himself, and we may look forward, under his *régime* at the Admiralty, to a continuance of the important part which the Royal Corps of Naval Constructors has played in the past history of our Institution. Mr. Watts carries with him to his new post the confidence and heartiest good wishes of his colleagues on the Council, and, I am sure I may add, of every Member and Associate.

The Board of Admiralty have, during the past year, shown an evident desire to encourage enquiry into various questions of importance which have come before them. Allusion has already been made to the exhaustive enquiry which has been conducted by the Boiler Committee under the able Chairmanship of Vice-Admiral Sir Compton Domvile, K.C.B., and time only permits of a passing reference to a few of the other Government or Departmental enquiries that have been instituted.

The Destroyer Committee, whose deliberations under Vice-Admiral Sir Harry Rawson, K.C.B., are not yet concluded; the Ship Subsidies Committee, under Mr. Evelyn Cecil; Mr. Arnold-Forster's enquiry into past delays in the construction of ships; and Vice-Admiral Fane's enquiry into the organisation and work of the Construction Department at the Admiralty.

The Navy Estimates for the coming financial year 1902–1903 are the heaviest that have ever been presented to the House of Commons, amounting in the gross to £32,376,717. There are, however, appropriations in aid which reduced the sum actually asked for to £31,255,500. The latter sum is £380,000 in excess of the corresponding amount for last year, and nearly nine millions more than that of the year 1897–98.

Whether the Estimates now brought forward, vast as they appear, are sufficient to provide and man a fleet equal to the needs of the Empire is a question upon which there is evidently a diversity of opinion; but, at any rate, we no longer hear that outburst of hostile criticism which accompanied the announcement of the Estimates some years ago.

The vote which most interests the members of the Institution of Naval Architects is doubtless the Shipbuilding Vote, No. 8. It is made up of three sections: 1, Personnel, 2, Matériel, and 3, Contract work. The two former sections refer to Government or dockyard operations; and in both we see decreases for the coming fianancial year on this year's estimates.* This falling off is, however, more than compensated for by the large increase in contract work, amounting as it does to not very far from a million (£980,300) on the figures of the present year. The large amount to be spent in the private shipyards of the country (£6,757,920 gross) will be more welcome to contractors, as the building of war vessels for foreign navies had rather fallen off of late. It is most desirable that, in time of peace, the art of warship construction should be extensively practised in the various building yards and engine shops of the country, so that we may be prepared for emergency should hostilities break out.

At the beginning of the financial year (April 1, 1902) there will be under construction—13 Battleships, 22 Armoured cruisers, 2 Second-class cruisers, 2 Third-class cruisers, 4 Sloops, 2 Auxiliary

* **The financial year commences April 1.**

Vessels, 10 Destroyers, 5 Torpedo Boats; and it is expected that during the coming year there will be completed and passed into the Fleet reserve—5 Battle Ships, 7 Armoured Cruisers, 2 Sloops, 2 Auxiliary vessels, 2 Destroyers.

It is proposed to commence during the financial year 1902–03—2 Battleships, 2 Armoured Cruisers, 2 Third-class Cruisers, 4 Scouts, 9 Destroyers, 4 Torpedo Boats, 4 Submarines.

A programme of re-construction has also been decided upon. It will include alteration in the armaments of the battleships of the Royal Sovereign class, and of the battleships *Barfleur* and *Centurion*. The large cruisers *Powerful* and *Terrible* and the *Arrogant* and *Talbot* classes, comprising 13 ships, will also have alterations made in their armament. The chief feature of this scheme consists in the additions made to the secondary armament, either by additional 6 in. guns or by substituting them for 4·7 in. guns.

A matter which will be of especial interest to Members of the Institution is that the Admiralty has decided to create a new class of vessel which will be known as the Scout class. They are apparently intended to be somewhat of the destroyer type, but with greater sea-keeping power so as to be more efficient for serving with the Fleet. As it is proposed not to initiate a design for the new class at the Admiralty, but to invite the private shipbuilders of the country to submit designs to fulfil certain conditions, no doubt members of the Institution will be closely identified with the new class.

A new departure has also been taken in making arrangements which will provide hospital ships for the purpose of accompanying Fleets on active service. As far as can be seen they will be called into existence when required, certain ships being earmarked for this purpose (belonging, I presume, to the Mercantile Marine), and they are having the necessary fittings prepared, to be put up when required.

I have now alluded shortly to the different questions of the day which appear of interest to this Institution, and I thank the members present for the manner in which they have listened to my address.

Gentlemen, we now come to what I think you will all agree, and I certainly think myself, is the pleasantest part of our duties to-day, and that is to make to our late most respected and most beloved Secretary, the presentation which was alluded to in the Report of the Council. I shall always, so long as I have the honour of being your President, congratulate myself on the fact that I was for some months, in the first year of my appointment, associated with Mr. Holmes. I can only say that I am afraid he will have to listen to a good deal of praise of himself before this presentation is finished, but I must say I cannot imagine a man more fitted for the post which he held in this Institution. We are fortunate in our present Secretary, who, I have no doubt, will in time rival our late one, but I feel perfectly certain that you will all agree with me in saying that no one could have shown greater courtesy, greater tact, greater judgment, or greater skill than Mr. Holmes did in the performance of his duties as Secretary of this Institution. Mr. Holmes gained the esteem and the respect of all with whom he was associated, and it will be long before his name is forgotten in this Institution. I am afraid I am one of those who never get used to the sound of their own voice, and I feel that I am not able to make a long speech upon this or any other subject; I can only do so when it is written down; but if I spoke for a week, I could not say more than I have already, to show the interest and appreciation we all feel, not only in the past career, but in the future of Mr. Holmes. He has gone to take up a very high and distinguished position in his native country, and we feel, from what we know of the manner in which he performed his duties in the past, that he will in the future receive the same appreciation in his new sphere that has hitherto been accorded to him. Therefore, gentlemen, without any further preface, I will ask Mr. Holmes to allow me to present to him this very handsome cup, which I hope he will be able for many years to see before him upon his dinner table. I have also to present him with an illuminated address which has been drawn up with the approval of the Council, and which, I hope, he will retain as a memento of his services to this Institution. And lastly, gentlemen, there is a cheque representing that sum of money which you have heard mentioned in the Report of Council. I heartily congratulate him on the proofs he has received of the esteem and regard in which he always has been, and always will be, held by the members of this Institution. (Prolonged applause.)

Mr. GEORGE HOLMES (Honorary Member and late Secretary, I.N.A.): My Lord Glasgow, my Lords, Ladies and Gentlemen, the last sentence of the much too generously-worded address which you have just presented to me makes me feel how intensely pleasant it is to be once more amongst so many old and true friends. Nothing in life can ever make up for the loss of one's friends. My Lord, I feel I have before me to-day certainly the most agreeable, but, at the same time, the most difficult, duty which it has been my lot to perform since first I became officially connected with this Institution. I do not know how I shall ever thank you for all that you have done for me, I feel deeply touched by your kindness and your generosity. You have done everything it was possible to do for a departing servant. You have conferred on me the highest honour in your power by electing me an honorary member of the Institution. There is nothing I value more in this world than that honour. Then you have presented me with this beautiful address, far too generously worded, and with this artistic cup, which, as our President says, will at all times, as long as I live, recall to me your kindness, and the many delightful occasions on which we have been associated in the past. And, lastly, the members, by their generosity, have made me a most handsome gift, which I really do not know how to thank them sufficiently for. No words of mine can adequately express my feelings to-day. My Lord, in your address, you have been pleased to refer to what you consider my services to the Institution. You have praised those services far above their deserts. The success of this Institution is not due to any efforts of mine, it is due to the fact that it fulfils a great national purpose. Who can think of England without her Mercantile Marine and her Royal Navy, without the ships that carry her commerce to the uttermost ends of the earth; that bring her Colonists—those pioneers of civilisation and justice and ordered liberty—to many a dark corner of the world, and bear back to us those proofs of support and of affection which have been so generously given to the Mother Country, during the last years of storm and stress? You are the men who render all this possible, and I have, therefore, always claimed for this Institution that it fulfils one of the greatest purposes in this country. Those purposes have been aided by a singularly wise and disinterested Council, presided over by a succession of most able Presidents, under each of whom I have had the honour to serve, and as to whose merits I therefore can testify better, perhaps, than most. They are men who have given their whole hearts and their best energies to the service of the Institution; all of them distinguished men who have served, and are serving their country, in various capacities, and in many parts of the world. All that I have done is to carry out their instructions to the best of my ability, and that has not been too well; but I have been aided in my efforts by a very devoted staff, whose merits I should like once more on this, the last occasion I shall have the opportunity of doing so, to place before the Institution. I would also, if I may be allowed to do so, congratulate the Institution on the choice the Council has made of its new Secretary. During the short time he has been in office, he has managed to endear himself to all who have come in contact with him, and to me it is an inexpressible delight and relief to know that the affairs of the Institution have been, and are being conducted so admirably. My Lord, once more I thank you all from my heart for what you have done for me. My connection with this Institution will always be to me the most agreeable recollection of my professional life. (Prolonged applause.)

The PRESIDENT: Gentlemen, the next duty which falls upon me is the presentation of the Gold Medal, which was awarded for last year to Professor W. E. Dalby, M.A. I think Professor Dalby is present, and I have much pleasure in presenting him with the Gold Medal. There is also a Premium of Books, which has been awarded to Commander Hovgaard, late of the Royal Danish Navy; but as he is not in this country, I am afraid we cannot present it to him in person. I will, however, take care that it is duly forwarded to him as the award of this Institution.

RESULT OF THE BALLOT FOR THE ELECTION OF MEMBERS AND ASSOCIATE MEMBERS OF COUNCIL.

Before the reading of the first paper on Thursday morning, the Scrutineers appointed to examine the ballot papers for the election of the new Council, presented the following report:—

Institution of Naval Architects,
5, Adelphi Terrace, London, W.C.,
March 19, 1902.

MY LORD,—We have counted the votes recorded on the balloting sheets enclosed in the ballot box given to us this day, and beg to report that the result of the counting is that the following gentlemen have been elected:—

As Members of Council—

DEADMAN, H. E.
BARNABY, S. W.
THEARLE, S. J. P.
SWAN, H. F.
DUNLOP, J. G.
THOMPSON, R.

As Associate Members of Council—

LEWES, V. B., PROFESSOR.
RILEY, JAMES.

We are, my Lord,

Your obedient servants,

(Signed) W. E. SMITH, *Member of Council.*
H. M. ROUNTHWAITE, *Member.*

To the Right Hon. the EARL OF GLASGOW, G.C.M.G., LL.D.,
President, Institution of Naval Architects.

TORPEDO-BOAT DESTROYERS.

By S. W. Barnaby, Esq., Member of Council.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 19, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

So much attention has been drawn to the Destroyer class of the British Navy during the past year, that no excuse is needed for a short paper on the subject.

It will necessarily be short, because no full treatment of the question is possible, in view of the fact that a Committee is now sitting at the Admiralty, and is investigating all the cases in which defects due to weakness of construction are reported to have developed during service at sea. I shall not, therefore, attempt to discuss these cases, because the facts have not been authoritatively made known. Neither do I wish to express an opinion as to the loss of the *Cobra*. I will only say that, although she was an exceptional and experimental vessel, still, if we are to believe that no injury to the bottom preceded the accident, it must be considered surprising that no preliminary straining or labouring of the joints gave warning of danger. We have no experience of mild steel, of the high quality to which we have become accustomed, having failed suddenly, even under repeated applications of stresses well within the elastic limit of the material. But as to what really happened, the evidence, to my mind, is not at all conclusive.

The problem which was originally set the Destroyer builders was, to produce a small vessel which would be faster than a torpedo boat, and would carry a heavier armament. These were the sole conditions imposed. Messrs. Yarrow and Messrs. Thornycroft, who made the first designs, naturally worked on the lines of the torpedo boats which both firms had been building for years. Speaking for Messrs. Thornycroft, we had confidence, from our experience with these boats, that if Destroyers were developed on the same lines they would be at least as seaworthy as torpedo boats, if not more so. We had built over 200, some of which had made voyages in all weathers and to all parts of the world, and not one had been lost at sea through insufficient strength. We considered that they must be amply strong enough to live through any weather in which they might be caught, but that the officers and men might reasonably be asked to submit to the same amount of discomfort at sea as was borne by the crews of torpedo boats, seeing that they were designed for the special purpose of catching these boats.

The torpedo gunboat of 800 or 1,000 tons is probably the smallest vessel in which any degree of comfort for the crew can be secured when it is necessary to remain constantly at sea; but although the torpedo gunboats may be able to overhaul torpedo boats in rough weather because of their size, a class was wanted which would overtake them in any weather, and I think this want has been met by the present Destroyers.

So far as I know, this condition of discomfort has been cheerfully accepted by the Navy, and the important question is, can these small vessels of high speed be made reasonably safe at sea? I think there is no doubt that they can.

The strains coming upon a ship among waves are not exactly, or even approximately calculable. The effect of the inertia forces produced by the rapid motion of the ends of the vessel as she pitches and scends cannot be estimated, because the velocity of the motion of the parts is not determinable. It is not necessary to enumerate the complex forces produced by the motion in a seaway, it is sufficiently evident that the data for exact calculation are altogether wanting. Then, again, while at one moment the deck of the vessel forms the top of the girder and the keel the bottom, at the next moment the rolling of the vessel may make the corner of the deck take the place of the top, and the opposite bilge that of the bottom of the girder.

All that it is possible to do is to establish a scale of comparison by which we may judge of the safety of one vessel by comparing her with another which has shown no sign of distress during the time that she has been at sea, and even then it is always possible that exceptional circumstances may occur, which may cause us to modify from time to time our standard of comparison.

It is usual to suppose, by considering the vessel first as poised upon the summit of a wave of her own length, and then as lying between the crests of a pair of such waves, and reducing the hull to the form of an equivalent girder, that a method of comparing the stresses to which ships are subjected at sea is possible.

The vessel may never be in such a condition, probably never is, and if she were, the stresses would doubtless be different from what they are calculated to be; but in default of a better way of ascertaining if a given ship stands at least as good a chance of safety at sea as some other ship which has proved to be satisfactory, the test is a valuable one. It is usual to assume the height of the wave for vessels of this size as about $\frac{1}{20}$ of the length, and some idea of the severity of the supposed conditions may be obtained, when it is stated that it means that a 210 foot Destroyer is immersed to the gunwale at the two ends, and that only about 2¾ feet of the depth of the hull amidships is in the water when she is lying in the trough, and that when poised on the crest, about 34 feet of each extremity of the ship is out of the water altogether.

It is, of course, of great importance that the deck should not only be sufficiently strong to avoid buckling locally under compressive strains, but that the form of the deck as a whole should be rigidly maintained. In the long machinery compartments, where there are no bulkheads, either transverse or longitudinal, great care needs to be taken that the deck beams and longitudinal deck girders are sufficiently strong for the purpose. In some of the latest Destroyers built abroad, where the engines are placed between the boilers, the coal bunker bulkheads are continuous through the engine-room, and greatly add to the strength amidships, but this arrangement obliges the engines to be placed one in front of the other, since there is not sufficient width for them to be abreast. This so greatly adds to the length of the ship, and consequently to the bending moment, that it is to my mind a doubtful advantage. The longitudinals and deep beams over these spaces will no doubt be reinforced in boats in which any tendency to bending has been observed. I think that they might with advantage be further supported by pillars, in order to strengthen the deck to withstand the compression to which it is subjected under a sagging moment.

Our practice has been to make the deck perfectly straight for the greater part of the length of the ship, giving no sheer except at the bow. Any sheer amidships prevents the deck from taking its proper share of the tension coming upon the upper works under hogging moments, and throws undue stress upon the sheer strake.

One of the most important factors determining the strength or weakness of a Destroyer—more important, in my opinion, than the thickness of the plating—is the ratio of depth to length.

In the *Albatross*, which is the longest we have built, there are 15¼ depths in the length; but in some of the class, I believe, the depth is much less in proportion. Two Destroyers may have the same scantlings of plates and frames, and be alike in length and displacement, and yet one may be a weak ship and the other a perfectly sound one; and especially may this be true if there is a great disproportion in relative depth.

As regards details, to which we attach great importance, I will only mention one or two. Unless the greatest care is taken in levelling the shell and deck plating by hand upon the slab, parts of the plate will be more severely strained than other parts, and the strength of the whole plate will thereby be reduced; but this is an operation requiring great skill and taking much time. No amount of machine rolling will level these plates as they require to be levelled, if the full advantage is to be taken of every pound of the material.

Joggling the butts of deck plating is, in my opinion, an undesirable practice, especially when this is of high tensile steel. I think it not only injures the plates to treat them in this way, but it also prevents the full tensile strength of the plate from coming into play. A joggled stringer butt, for instance, is subjected to an unfair pull, which has to straighten the kink in the plate before it can stretch it, and an excessive strain is thrown on the sheer strake.

The riveting is another detail requiring the greatest care, and we never allow this to be done by piece-work. The introduction of high tensile steel has made the question of riveting of still greater importance. The value of this material would be much enhanced if rivets of the same strength could be employed; but there are difficulties attending its use, and not much experience has been gained with hard steel rivets up to the present.

If rivets, either of mild or hard steel, are riveted cold, they appear to become brittle and treacherous, and our experience has satisfied us that good Lowmoor or charcoal iron rivets are more reliable, and as strong as steel rivets when put in cold. Rivets of small diameter cannot, we believe, be safely put in hot, whether they are iron or steel; because they are liable to waste by scaling in heating, and are cooled so rapidly on being put into the holes, that there is not time to properly knock them up before they arrive at a temperature at which hammering is injurious to steel. We therefore prefer iron rivets for sizes up to $\frac{3}{8}$ in. or $\frac{7}{16}$ in., but those of $\frac{1}{2}$ in. or upwards might be of steel, and should be worked hot. It may be possible to use high tensile steel for these, if it is certain that its quality is not affected by the heating, on which point more experiments are desirable. The spacing of these rivets could be greater than with iron rivets, and the plate would be stronger at the joint.

The ratio of weight of structural hull to total displacement in Destroyers is not unduly light, and does not compare unfavourably with that found in other classes of warships. For example, Sir William White, in his "Manual of Naval Architecture," gives the weight of material contributing to structural strength in a steel-built first-class battleship of the present day as 18 per cent. of the total displacement, and for a typical swift protected cruiser of high speed, large coal supply, and heavily armed, as 20½ per cent. of the total displacement. This is considerably less than the percentage of weight of material contributing to structural strength in torpedo boats and torpedo-boat destroyers.

Although the "Thornycroft" Destroyers are not longer in proportion to depth than the earlier torpedo boats, we have taken more care to preserve the continuity of longitudinal strength than was done in those boats. Continuous keelsons, side stringers, and deck stringers have been introduced which were not fitted in the torpedo boats, and far more attention is now paid to the fitting of doubling plates in way of openings in the deck, such as funnels, fan cowls, hatchways, &c., in order to compensate for the material cut away by these openings, and thus to bring the strength there up to that of a normal section taken through the rivet holes at a frame, which should be the weakest section in the ship. To this question of compensation the Admiralty have very properly attached great importance in all their recent specifications, and there is no doubt that it is much more necessary to pay careful attention to it than it was in torpedo boats, on account of the increase in dimensions.

The longitudinal bending moments for similar ships on similar waves vary as the fourth power of the linear dimensions, so that the stress per square inch of material will increase with increase of dimensions if weight of hull vary as displacement, and as a rule this is found to be the case. Large ships are usually more highly stressed than small ones. M. Normand and others have shown that structural weights should vary as the four-thirds power of the displacement for equal stresses under longitudinal bending in similar vessels. But in dealing with moderate increases of dimensions, as in passing from a torpedo boat to a Destroyer, there are a good many of the scantlings, such as plating over propellers, doublings and chafing plates in way of anchors, coal bunker bulkheads and shovelling flats, and other parts which do not count for much in structural strength, but which require to be of a certain minimum thickness for local strength in the smaller vessel, and which do not need to be increased in the same proportion as the rest. The weight thus economised can be utilised in thickening the deck, keel, and sheer-strake amidships above their proper proportion, and thus the stresses per square inch of material do not rise at the rate that they would otherwise do. As a matter of fact, we have found that by improvements in structural detail, such as I have mentioned, it has been possible to keep the estimated stresses in a seaway down to a figure which allows a good factor of safety, taking into consideration the strength of the high tensile steel employed.

There is no reason why boats having speeds of 30 or 31 knots at light draught should not be as capable of living through bad weather as a torpedo boat. This was the original standard, and provided that they are equally well proportioned and equally well built, they should run no greater risks than their prototypes did. If a higher standard of strength than this is now considered desirable for Destroyers for the British Navy, so that instead of working from a base they may always accompany the Fleet at sea, I believe this requirement can be met without a great sacrifice of speed.

It would be unfortunate if the exaggerated impression which has got abroad as to the frailty of Destroyers as a class should lead to a swing of the pendulum in the direction of increase of weight which should greatly exceed the necessities of the case.

If we go to such heavy scantlings, or if we so increase the dimensions in order to secure comfort at sea, that Destroyers can no longer be fast enough to overtake torpedo boats in smooth water, would not their usefulness be much impaired? Other nations will probably continue to build small fast craft of this description in which comfort is sacrificed to speed and efficiency, and can we afford to be left behind?

The present Destroyers, like the torpedo boats, are lightly built at the ends, and it is necessary to give special attention to the bow plating and framing in new boats, so that they may be able to maintain speed in rough water. Besides the local strengthening of the bows and a moderate increase of scantlings generally, say from ten to fifteen per cent., to enable them to stand more knocking about, I think it would be wise to increase the ratio of depth to length even above that of the *Albatross*. The strength should then be ample for all requirements.

I do not feel at liberty to say anything about the latest designs that have been called for by the Admiralty, in which builders have been left as usual a fairly free hand. The moderate speed specified, 25½ knots, is due chiefly to the conditions of trial, which has now to be made with full load on board, and not, as previously, with a very light load. Although I should have preferred to see the trial made under average conditions of load, that is, in fully equipped condition, but with bunkers half full rather than in either of the extremes of loading, still I can see no reason why thoroughly good boats should not be built under the new conditions laid down, provided that moderate views are allowed to prevail as to the hull weights and dimensions which are left to the judgment of the designer.

They will not be as fast at a light draught of water as the present boats are in that condition, but their speed will be increasing all the time as the coal burns out, and the average speed should be considerably more than that obtained on trial.

In view of recent events, what is a safe stress upon the material either of the deck in compression or upon the keel in tension? Do our views upon the subject require modification, and, if so, to what extent?

Fairbairn found, many years ago, that the joints of an iron-riveted girder sustained upwards of three million changes of one-fourth the weight that would break it, without any apparent injury to its powers of ultimate resistance. It broke, however, with 313,000 additional changes when loaded with one-third the breaking weight, evidently showing, he says, that "the construction is not safe when tested with alternate changes of a load equivalent to one-third the weight that would break it." There are numbers of large ships at sea in which the stresses must be considered very high if judged by this standard, but which have shown no signs of distress. This would seem to indicate that the extreme conditions assumed in the stress calculations are very rarely met with, and that if they do occur they last for a comparatively short time.

But we have to remember that it is more difficult to get a thin plate to stand a compressive strain than a thick one, and also that small vessels are likely to encounter waves which will strain them more frequently than large vessels. The waves which are assumed in stress calculations of battleships are given by Sir William White as 380 ft. in length, and 24 ft. in height; while those assumed for a 210 ft. Destroyer are 10½ ft. high only. Of course, the Destroyer which has to keep company with a battleship may be called upon to encounter the 24 ft. waves; but these, on account of their greater length, do not produce such a severe bending moment as the smaller wave. The boat cannot stretch from crest to crest, and is better supported than upon the shorter waves of less height.

But, as I said before, these calculations cannot be depended upon for exact figures, and are only useful as methods of comparison.

I have my own opinion as to how high a stress it is safe to allow; but it would serve no useful purpose to attempt to fix a definite limit, where so much depends upon workmanship and other indeterminable quantities. The builder's own experience should be his guide. If all vessels had to be built to the rules of insurance societies, there might be less risk, but there would certainly be less progress; and torpedo boats and Destroyers could never have come into existence.

DISCUSSION.

Mr. A. F. Yarrow (Vice-President): My Lord and Gentlemen, I have much pleasure in complimenting Mr. Barnaby on his very interesting paper; not only is the paper of great value, but the subject is one which at present is attracting much attention. I think, in the construction of these light vessels, the elasticity of the structure is one of the most important points to bear in mind. A great deal of the success of these light structures is undoubtedly due to the fact that under blows of the sea, and strains due to waves, they give somewhat, and if it were not for that springing they would be destroyed. I think the question of elasticity cannot be too much studied. Of course, elasticity in the structure must be, as far as possible, uniform. Destroyers in rough weather twist and bend, and, if the movement be concentrated locally, if the bending takes place mainly in certain parts and is not distributed more or less throughout the length, the metal in that part of the hull where the bending mainly takes place gets fatigued, and sooner or later comes to grief. One ought, therefore, to look upon the structure as an elastic body and deal with it as such. Not only will a reduction in section produce a weak place, but an increase in section will also do the same; for example, if we made a hole in a clock spring, it would probably break very soon at the hole. In the same way, if we strengthen a clock spring over a portion of its length, it will give way at the place where the section changes; so we have to add material very carefully in a structure of this kind. There is no doubt that the buckling strains to which the deck is subjected are among the most important things to be studied. In relation to this, one must bear in mind that good workmanship here plays a much more important part than it does in most parts of a vessel, for this reason: if a plate is not "fair" and a tensile strain comes upon it, it gradually gets pulled into such a form that it can stand the strain better than in its original form; while if a plate which is out of shape or "unfair" has to stand a compressive strain and begins to buckle, then, as buckling proceeds, the plate becomes weaker and weaker to withstand the further buckling strains; and it is, therefore, of the utmost importance, when we are dealing with light plates subject to buckling strains, that the workmanship should be beyond question. With reference to the butts of the deck, our custom is for all butts, both to the deck plating and the sheer, to put a butt strap inside and out. I believe that is the only fair way of dealing with light plates subject to buckling strains, because the strain passes through, as it were, the centre of the plate and through the centre of the joint, which it does not do in the case of a single butt strap. I had no idea that anybody would join the deck butts by joggling, and really, I do not think that anyone who does that has fully appreciated what he is doing, because in that case you are, as it were, just starting a buckle. If one wanted a plate to buckle, one could not do better than make it originally in that form, for we know that when a tensile strain comes upon a bar or a plate which is nicked, it is pretty sure to break at the nick, and it is a very dangerous thing to allow a nick in anything subject to tension. A small buckle is equally dangerous, as it is the commencement of a big buckle in any light plate subject to compression. As I say, our practice is that all the deck plating should be joined by a butt strap on both sides, and the same system applies to the sheer strake as well. I am inclined to think that that is the only reliable mode of keeping the plate in form, because if you have a butt strap on one side only, that has a tendency to start a buckle, for in a light plate one cannot reckon on having the butts so that they touch; that is practically an impossibility. It is in the details of construction that one cannot be too careful to avoid the tendency to cause a buckle, and I am glad Mr. Barnaby has drawn attention to the great importance of flattening the plates before they are bent into shape. If you have a plate which, when it comes from the works, has a buckle in it, that buckle must be taken out, because, although the plate may appear flat, the buckle will show itself when a compressive strain comes upon it. Similarly, we attach very little value to the plating over the boilers, *i.e.*, the centre portion of the deck, which passes over the steam chests, owing to the heat from the latter expanding the plate immediately above it. We all know that in walking over the deck (if it is a thinly plated boat) those plates, just where the boiler comes, begin to buckle before you reach them, and they are, therefore, not in a good condition to take any compressive strain. In fact, I think that those plates are actually doing harm, because they are in reality the means of starting a buckle. Mr. Barnaby referred to insurance companies, and it may be of interest to members to know that for destroyers built in different parts of the world, the rates of premium vary from 6s. 8d. to two guineas. I do not say that the risk is in proportion to that, but it is an indication of what those people think who make a study of that particular subject. In conclusion, I am very glad of the opportunity of expressing my firm conviction that torpedo-boat destroyers with the present scantlings are of ample strength to ensure the safety of those on board in any weather, if skilfully handled, and provided the workmanship is exceptionally good, and that proper attention is given in the design to those details which are so essential in vessels of light construction. Any further material put into the structure of the hull would only tend to reduce the speed and usefulness of this class of vessel without any corresponding advantage.

Mr. J. I. Thornycroft, LL.D., F.R.S. (Vice-President): As to the larger aspect of the question of seaworthiness, and of suitability for the particular purposes for which these vessels are designed, it must not be forgotten—though Mr. Barnaby really has already laid stress on this point—that it is of the very greatest importance that the builders should not be so handicapped as to make them turn out vessels which will not answer the purpose for which they were originally designed, and it must be conceded that more risk is allowable in this kind of vessel than in an ordinary one. I am very glad to hear how much confidence Mr. Yarrow has with regard to what can be done, and that he thinks these boats can be made successful in resisting the action of the sea; but it must be always allowed that there are circumstances in the behaviour of waves at sea, for instance, which do not conform to the beautiful theoretical curve of sines or anything like it, and these vessels, which are necessarily very light, may be subjected on occasions to something they cannot withstand; but if they are built so as to have a fair chance of safety, I think all is done that can be asked for. It is of the greatest importance that these destroyers should not be so legislated for as to really put them out of existence. I understand also that this class of vessel has been very valuable for the training of our sailors, and in forming a school in which men can take a responsibility which it is not possible to give them in large ships; young officers can learn to handle these vessels—and difficult they are to handle, requiring, as they do, a very quick appreciation of the surrounding circumstances—and in that way they serve a very useful purpose as a training school for our sailors.

Admiral Sir Nathaniel Bowden Smith, K.C.B. (Associate): My Lord President and Gentlemen, I would much rather have heard the opinions of some of the experts here about these boats, but, as I have been called upon to speak, I will venture to offer a few remarks on the paper we have just heard, disclaiming for my part any knowledge of the building of either destroyers or torpedo boats, nor can I claim personally to have any knowledge of handling them. This type of craft came into general use in our service some time after I had arrived at a position on the list which precluded me from serving in them, but from my recent experience when in command at the Nore, I came to the conclusion that our destroyers were too frail for the purpose for which they were required. As you all know, we have at each of the three home ports, Portsmouth, Plymouth, and the Nore, a flotilla of eight destroyers, which are kept in constant use for exercise and practice, and most valuable they are for the instruction of both officers and men, as Mr. Thornycroft has just pointed out. But I must say that after a three weeks' cruise they hardly ever returned into port without one or more of them being more or less damaged in the hull. They are so very slightly built that the least collision between two of them, the rubbing against a buoy or floating wreckage when coming into harbour after dark, or the slightest impingement against a wharf or jetty is bound to cause serious damage. I quite understand the builders wishing to keep down the weights as far as possible, but I am glad that the Admiralty has appointed a Committee to inquire into the matter, which, I believe, will lead to the strengthening of some of our present boats and to the designing of new ones of somewhat stouter build. In the old days, if we were opposed by a gale or heavy weather, we always lay to. Under the present conditions, with the enormous power we have in vessels, it is sometimes the fashion to push them, perhaps unwisely, and thereby subject them to enormous strains. In former days, we did not lay by merely to save our boats and avoid damage, but also because we could not well do otherwise. Long after steam was introduced into warships, it was of such small power that we could make no progress against a strong head wind, and the only thing we could do was to lay to, and drift as little as possible. The author alluded to the unfortunate loss of the *Cobra*. She was at the time, I believe, making for port. If she had been lying to with her head off shore and bow to the waves, I suppose no harm would have happened to her. In making that suggestion, I do not wish to impute the slightest blame to the unfortunate officer in command or to anybody else, nor would I say anything to discourage young officers from occasionally pressing their boats in bad weather, because they might have to do so in war time, nor do I imply that these vessels are not perfectly seaworthy. The author tells us that Thornycroft's firm have sent out some two hundred destroyers to different parts of the world. I remember in 1881 or 1882, when out on the south-east coast of America, some torpedo boats—not destroyers—coming out to the Argentine Government. They had their screws taken off and some of their weights removed and came out under sail, and I remember very well, when the first boat appeared in the roadstead at Monte Video Harbour, we were all wondering what the strange little craft could be. I should not have addressed this assembly if I did not wish just to emphasise this fact: that these vessels, being very slight as regards their frames and plates, are furnished with engines and boilers of enormous power, and, if pressed against a heavy sea, they must be subjected to an enormous strain, which the author himself tells us "cannot be even approximately calculated."

Mr. Francis Elgar, LL.D., F.R.S. (Vice-President): My Lord and Gentlemen, I fear there is but little I could add with advantage to what has already been said on this subject, but I would be pleased to submit to the meeting some opinions I have formed during the last few years in connection with the design and building of these boats. Mr. Barnaby has expressed very clearly in his paper—as clearly as he is well able to do under the conditions imposed upon him, not being, of course, free to publish details of design and construction—what is the general nature of the problems that have to be dealt with in the structural design of destroyers; and, if I might express such an opinion, I think he has performed the task very judiciously. We find, as designers and builders of torpedo-boat destroyers, that nothing of structural weakness is heard from the Admiralty after these boats leave our hands. My firm has been building them for seven or eight years, and we have had no complaint whatever, official or otherwise, of structural weakness having shown itself in any case. It is only within the last few months that we have heard doubts expressed in various quarters, and these have been of a general and somewhat vague character, as to the strength and seaworthiness of torpedo-boat destroyers as a class. Now it must be remembered, in connection with this class of boat, that when they were first built they were designed for the special work of chasing and running down ordinary torpedo boats; everything else had to be sacrificed to speed, so long as they were made sufficiently seaworthy for following up and coping with ordinary torpedo boats at sea. The designers of the torpedo-boat destroyers were not asked to produce boats that would, before everything else, be able to keep the sea at all times and in all weathers, and to perform any general duties at sea that might be put upon them. Had that been so, I think some of us might have hesitated before designing the boats exactly as they now are. They were intended, or so we believed, to work within a limited range from a base, and to keep torpedo boats away. I suppose the ideas in the Navy were necessarily rather indefinite at the time as to the precise extent to which it might be found desirable to employ such boats, but the Admiralty, after getting possession of them, have naturally tested very thoroughly the extent to which they can be used at sea. I, for one, have been surprised to find that these boats are as efficient and safe at sea as they have proved themselves to be. A considerable number of torpedo-boat destroyers have been sent to all parts of the world, and have been severely tested upon long voyages in bad weather; and I think it is wonderful that they have come out so well as they have done. I doubt if any designer would have had the courage to guarantee, merely upon the results of his calculations, that these boats would have stood the straining action of the sea under various conditions so well as they have generally done. I quite agree with the view Mr. Barnaby expresses in the paper, that the data obtained by calculation are very unreliable by themselves as a standard for determining the value of these boats at sea, and that it is not safe to judge of their structural strength merely by the results of calculation.

It is only practical experience, and exhaustive trials of the boats under working conditions at sea, that can show exactly what their qualities are, and furnish a real criterion of their value for sea-going purposes. I consider that the results up to date, taking them all round, have shown that, judged by that criterion, they have come out very well. So far as I have heard anything about defects due to structural weakness, they have been pretty well confined to the top of the boats—to the deck—and they have generally appeared to be connected with or caused by the buckling of the deck plating between the beams, or the failure of the deck plating at a butt joint. I am surprised, with Mr. Yarrow, to see a butt with a joggled lap introduced into this class of work, where the greatest obtainable strength and efficiency is required from every item of material, and it is especially surprising to find this in a part of the structure where tensile strength is considered so important that specially high tensile steel is substituted for the ordinary ship steel. This is obviously a weak form of butt joint, and not one that is adapted for bearing great tensile or compressive stresses. Now, although the defects of which we have heard have generally been in the decks of these boats, the calculations of structural strength indicate (so far as I have been able to go into them) that it is not really the top of the boat that is the weakest part, so far as the distribution of the material over the transverse section is concerned. I am inclined to think that the bottom of the boat is, on the whole, weaker structurally than the top. The reason that the decks have failed in some cases does not appear to be because there is not material enough in the deck, but because it has not been sufficiently stiffened and supported against buckling. The plating has apparently failed by buckling long before the full amount of stress it could otherwise have borne came upon it. What the calculations seem to point to is that if it were considered or found necessary to increase the general structural strength of these boats, the way to do it would not be to add more material to the top, but to put it into the bottom ; at least, if greater weight of material is found to be required. I think, however, it is very doubtful if more material is necessary. Great attention is required to the butt connection of the plating throughout and to the decks, so as to prevent buckling of the plates and enable the deck plating to exert itself to the full extent of its strength in resisting the stresses that come upon it. There is one other question that seems to require attention, and that is, whether the riveting might not be increased with advantage where high tensile steel is employed. I doubt if the rivet connections of the butts, for instance, are sufficient, at present, to enable as much as possible of the full structural value to be obtained out of the material. If any change were to be made in this respect, the double strapped butt, as shown by Mr. Yarrow, is one which would, of course, enable additional value to be obtained out of the rivets. I will not occupy time with any further remarks, but conclude by saying that I thoroughly agree with Mr. Barnaby and Mr. Yarrow as to the necessity of paying the greatest attention to details in every portion of the structure of these boats. I am satisfied that by so doing, by stiffening the plating in every part of the structure, and supporting it so as to ensure that every part will do its proper work and give full value in structural strength for the weight it adds to the boat, and by careful attention to every other particular and detail, however small it may appear to be, the strength of these boats can be improved, and more real good can be done than by making the plating thicker, stronger, and heavier throughout, and thus handicapping them with unnecessary weight and risking a failure to fulfil in a thoroughly complete and satisfactory manner the main objects for which they were introduced into the Navy.

Mr. James Hamilton (Member of Council): My Lord and Gentlemen, I wish to say that it occurred to me, on hearing this paper read, that, in the opening paragraphs, Mr. Barnaby rather suggested the idea that he is giving way too much to the popular idea that there can be no useful warship or vessel that does not fulfil the condition of being strong enough to be able to keep the sea with a full measure of safety in all weathers. I say a full measure of safety, for that is the key to the whole matter. Now I think, my Lord, that there are many commercial vessels, Channel vessels and others, that scarcely fulfil that condition. Every ship is strong enough for some work, just in the same way as every engine is strong enough for some power. We can hardly expect that every ship should be built to do the same work, any more than we can expect that every crane shall be designed to carry the same load. There are such things as factors of safety widely different as between ships and cranes and the like, and it seems to me that what is wanted is to classify vessels, and let the public and the officers know what is the safe working condition of every warship and every vessel; and when deeds of derring-do have to be done, let them understand what are the additional risks that they have to run. By all means, let them take those risks, but let it be understood what they are.

Admiral the Hon. Sir EDMUND FREMANTLE, G.C.B., C.M.G. (Associate): My Lord, with your permission I should like to say a few words, principally because I am in the unusual position of differing somewhat from my friend Sir Nathaniel Bowden Smith, and, I am afraid, from Naval officers generally. In this instance, with regard to the strength of destroyers—and I will not go at all into the question of their construction, which I am not really competent to discuss—I will say this: that my sympathies are entirely with the constructors of these destroyers, and with the constructors of torpedo boats—with Messrs. Thornycroft and Yarrow, and Laird and others, who have built them so successfully in so far as speed is concerned, and not so much with the Naval officers who have tried them. I entirely agree in their value as instructional vessels for our young officers and for our men. I believe they are of the greatest value to the Service, and I am particularly struck by what Mr. Barnaby says in his paper:—"The problem which was orginally set the destroyer builders was to produce a small vessel which would be faster than a torpedo boat and would carry a heavier armament. These were the sole conditions imposed. Messrs. Yarrow and Messrs. Thornycroft, who made the first designs, naturally worked on the lines of the torpedo boats which we had been building for years. Speaking for Messrs. Thornycroft, we had confidence, from our experience with these boats, that if destroyers were developed on the same lines, they would be at least as seaworthy as torpedo boats, if not more so." Now a great many of us (and certainly I am one) are old enough to recollect when we first had torpedo boats. We sent them to sea, in 1885 I think it was, to try them under all conditions, the idea of the French especially being that they could keep company with the line-of-battle ships in all weathers, and that they would be a very useful adjunct to the fleet at sea. In 1885 they were tried under one of our most distinguished officers, Admiral Sir Geoffrey Hornby, and there is a very scathing criticism of their condition at the end of that time by an American officer, Commander Bainbridge Hoff, who spoke of them as having the appearance of "famished beasts," and who said that they were all knocked to pieces and were absolutely unfit to do service under those conditions. Now the same thing, to some extent, has been the case with the destroyers. We have over-tried them. They are capable of great exertions, and of going at great speed if carefully nursed. I am aware that when I make use of the expression "nursed," a Naval officer would naturally say, "Oh, you want to keep them very comfortable and put them in a glass case!" but I hold very strongly that every vessel, like every horse, has its proper work to do, and that if you try it in a way that was not intended for it, you will naturally lose the benefit of its services. If you put a racehorse to drag a cart, you will very soon find you have no longer got a racehorse, and I really think that is the gist of the whole question. I do hope that the Admiralty will not go to the extreme of trying to make the torpedo-boat destroyer (which is built for extreme speed) capable of keeping the sea in all weathers, because under those circumstances, I think we shall probably lose our destroyer altogether, and we shall find we have a vessel which is a very inferior sort of cruiser.

Admiral Sir DIGBY MORANT, K.C.B. (Associate): My Lord and Gentlemen, I have no experience myself, but I have a son who is commanding a torpedo-boat destroyer, and who has now been twelve months in command of the *Quail*. He was in a heavy gale going up from the St. Lawrence to Halifax last autumn, and the cruiser *Proserpine* was convoying him. When they got into Halifax they found, on comparing notes, that they had made better weather of it than the *Proserpine*. Last autumn, they came down from Halifax to Bermuda in a gale of wind the whole way. They were also convoyed at that time, and I heard from the officer who was in command of the convoying vessel, that they were very uncomfortable, but that our boat was seaworthy. Shortly after, certain questions were sent out, and amongst them was whether my son wished to remain in the destroyer, and in his letter to me he wrote, "Of course, I said: Certainly, yes!" I thought I would just communicate this little piece of experience as a proof that late events have not caused any want of confidence in the seaworthiness of the destroyers.

Rear-Admiral W. H. Henderson (Associate): My Lord and Gentlemen, I only wish to say two words: one is to concur cordially with the remarks of Sir Edmund Fremantle, and the other is to note that our naval history shows that we have always had "flotilla craft," that is, small craft with smooth-water speed, but sea-keeping limitations. In introducing the powerful and fast flotilla craft which we call destroyers, we have not, I think, in the Service, thoroughly appreciated their limitations. I believe that with some minor structural improvements they are capable, under judicious management, of keeping the sea in heavy weather. I know that two destroyers in China, one, I think, the *Fame*, and the other the *Taku*, which had been taken from the Chinese and had been originally built in Germany, were caught outside Hong Kong in that very heavy typhoon which did so much damage there a year or more ago. They were nearly at the end of their coal supply, and were on a lee shore, but they managed to claw off the coast. Their chief difficulty was due to leakage at the bows, caused by the heavy plunging. This leakage was so great, that it was as much as they could do to keep it under.

Mr. Sydney Barnaby (Member of Council): My Lord and Gentlemen, I am very gratified indeed at the discussion which this paper has called forth, and especially by the views which have been expressed by so many Naval officers. I think all—yes, all—have expressed opinions that may be considered as quite re-assuring from the point of view of the safety of these vessels; in fact, it is only Admiral Sir Nathaniel Bowden Smith who thinks that they are not quite strong enough for the work which they should be able to perform, although he does not say, and probably does not think, that they are too weak for the work which they were originally intended to do. I quite agree with what Mr. Yarrow has said, that the boats which are well and carefully built, are now strong enough to stand any weather to which they may be subjected, because this very bad weather does not last long. You heard the statement as to the extraordinary number of changes of stress which these riveted girders will stand under a very heavy load, and to get a million variations of stress in a seaway would mean a very prolonged storm indeed. But if they are to be forced through a heavy sea at high speed, you can see immediately, by looking at the manner in which this destroyer (see Plate A) is now plunging her head into a very large wave, that the straining upon the fore part of the ship is very severe, and no doubt accounts for the leakage of the plates which is found there when the vessels are pressed at high speed through such weather. If they can be eased down, I think that leakage need not necessarily occur at all. As to collisions, of course, it becomes a very difficult thing to do more than by water-tight subdivision to enable vessels to stand a collision—even a slight one. I think that point ought hardly to be considered in connection with a destroyer. Dr. Elgar has very properly said that he thinks the bottom plating is more severely stressed than the deck plating. I am quite of that opinion. So far as the calculated stresses go, they are, as a rule, severer upon the keel than they are upon the deck, but the deck is so much less calculated to stand the compressive strains, as he has said, that it is the deck to which attention requires specially to be directed. As to the question of the factor of safety which Mr. Hamilton has mentioned, if you are going to legislate for bad workmanship you want a high factor of safety. If you are going to make up your mind to build the boats with the most perfect workmanship that can be put into them, then you can do with a very much lower factor of safety than would be otherwise necessary. As to the question of speed, that very distinguished torpedo-boat builder, Monsieur Normand, has lately produced two torpedo-boats which have given a speed on trial of thirty-one and a half knots. That is a very remarkable performance, and if destroyers are deliberately over-weighted to give them that amount of security at sea which some people would desire, I think all prospect of being able to catch high-speed torpedo boats of that sort must be given up, and then what is the destroyer to do—what is to be her function ?

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G.): Gentlemen, I think you will all agree with me that we have had a most interesting and fascinating paper from Mr. Barnaby, and considering the interest which is at present being manifested in this class of vessel, I am glad his paper has been the first upon our list, and I ask you to join with me in giving Mr. Barnaby a hearty vote of thanks.

The following written contribution has been received from Mr. C. HUMPHREY WINGFIELD (Member) :— "The important subject of the adequacy of the destroyers in His Majesty's Fleet to fulfil the purpose of their existence, and to withstand rough weather without undue structural stresses being experienced, is one which has of late somewhat exercised the public mind, and it is very satisfactory to find that an acknowledged expert like Mr. Barnaby, who is so thoroughly conversant with the subject, is able to assure us that, in competent hands, there is no reason why these vessels should not be so designed as to be capable of living through as bad weather as a torpedo-boat. Of course, the calculation of stresses in a vessel is at best only an approximation, and has to be combined with a " factor of common sense." This correction can only be made as the result of special experience, as indeed the author points out, the lack of which cannot be compensated for, when designing such craft as destroyers, by a knowledge of current practice in the design of larger vessels. The information given by the author is of greater value on this account, and he rightly urges the importance of keeping the deck of such a form as to avoid buckling stresses. He prefers making it straight, and points out that the greatest care is necessary in levelling the plates in order to avoid local strains. It has often struck me that the curvature purposely introduced in the hull plates between the frames in order to follow the curves of a ship model, is so much greater than the small buckles referred to in the paper, as to materially reduce the power of some of these plates to resist end compression or tension: for they would deform sideways with a smaller endwise stress than the straighter plates near the keel or on the deck. If so, I think these latter plates will not be relieved to the same extent as would be indicated by the " equivalent beam " method of calculation, and it is just here that the special experience possessed by Mr. Barnaby enables him to compensate so successfully for what I suggest might—with thin plates—be a very misleading method of ascertaining the stresses. Of course, if this is so, the nominal stresses will require a good deal of correction before the questions put by the author on page 6 can be definitely answered. His suggestion that depth is of great importance appears to confirm what I have said as to the difficulty of estimating how much of the total bending moments is taken by the curved plates which form so great a proportion of the structure. When put far apart, and kept from buckling in the way suggested by the author, the comparatively straight plates of the deck and lower part of the vessel are in a favourable condition to take, if necessary, a larger share of the total stress than would be indicated by the usual methods of calculation."

The following is the reply by Mr. S. W. BARNABY :—Mr. Wingfield is no doubt correct in stating that all the plates of a ship's hull are not equally well situated to resist the stresses to which they are assumed to be exposed. This is one among many reasons why the results obtained by the equivalent girder method of calculation are only useful for comparing the strength of one vessel with another not very dissimilar in form, and which has successfully stood the test of service.

ON THE STRESSES IN A SHIP'S BOTTOM PLATING DUE TO WATER PRESSURE.

By Mons. Ivan G. Boobnoff, Naval Architect, I.R.N., Member.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 19, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

I do not know of any question in the theory of elasticity which should interest the naval architect to the same extent as that of the flexion of thin plating. Indeed, the whole ship, from keel to upper deck, consists of plates, which are to fulfil the most varied purposes and to withstand all kinds of stresses. Owing to this, naval architects cannot be satisfied with approximate and rough practical formulæ, which may be regarded as sufficient by engineers of other branches of the engineering profession, and they are bound to examine and solve this question in detail. The series of communications read before this Institution, by such authorities as Professor G. H. Bryan,* Dr. F. Elgar,† Mr. T. C. Read,‡ Mr. J. A. Yates,§ and others, shows how pressing is the demand for the solution of this question, and this circumstance encourages me to submit my attempt to generalise some of the results already obtained.

§ 1.—The outer and inner bottom plating and the bulkhead plates of a ship present, in the majority of cases, a series of perfectly flat plates fixed to a rectangular network and subjected to water pressure, which may be considered uniform for each separate rectangular plate. Under this pressure the particular plate bends, and the chief problem for the naval architect consists in establishing the connection between the size of the plate, the amount of pressure, and the stresses which are caused by it.

This question, notwithstanding its apparent simplicity, has not been hitherto solved by the theory of elasticity under the conditions which correspond to its practical

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* On the Theory of Thin Plating. Transactions of the Institution of Naval Architects, 1894.

† Some Considerations relating to the Strength of Bulkheads. Idem., 1893.

‡ On the Strength of Bulkheads. Idem., 1886.

§ The Internal Stresses in Steel Plating due to Water Pressure. Idem., 1891.

realisation; in the practice of naval architecture, it is as yet only solved by the application of a roughly empirical method of comparison, and for the Mercantile Marine by the use of different rules devised by Lloyd's and the Bureau Veritas, without any theoretical explanation of these rules.

All the attempts that I know of to solve this question theoretically treat the plate as possessing either absolute stiffness or absolute flexibility. In the first case, its breaking takes place at a considerably greater load than that determined by theory; the formulæ which are deduced on the second assumption are better confirmed by experience, but the incorrectness of this hypothesis itself is obvious.

I shall not ascribe to the plate properties which it evidently does not possess (*i.e.*, absolute stiffness or absolute flexibility); therefore, the results which are thus obtained will, perhaps, be considered as more trustworthy than those previously obtained. Although I am unable to offer a perfectly exact solution, giving a strictly definite result, I will try—

(1) To establish the limits within which the tension on a flat plate, of given dimensions, loaded with a certain pressure, is included;

(2) To apply the deductions so obtained to the constructions used in the Mercantile Marine and in ships of war.

These results I have the honour to submit to the judgment of this Institution.

§ 2.—A plate fixed on its boundary resists external pressure partly by its stiffness, partly by its stretching. There is a marked difference between the laws of flexion of a stiff beam and the resistance of a perfectly flexible but elastic chain. We shall establish in the first instance these two limiting cases.

Let us consider a horizontal steel beam, of a uniform section, supported at both ends, the span being twenty to forty times the depth of the beam. If this beam is loaded with a weight uniformly distributed over its length, and we measure the strain, or the deflection, produced at any point of the beam, we shall observe that, until the load has reached a certain limit, the deflections vary proportionally to the load; and, as it is proved in the elementary theory of elasticity, the stress at every point varies in the same ratio. On a further increase of the load, the deflection begins to vary more quickly than the load, and in the case of all sound constructions, such as bridges or buildings, this limit of strain is never attained; but the stress at each point of the beam varies almost proportionally to the load, until the beam breaks at its weakest part.

Thus, if the breaking takes place under a load of, say, 6 tons, the ultimate tensile strength for the material of the beam being 30 tons per square inch, we may assert that under a load of 1 ton the greatest tension will not exceed 5 tons per square inch, and the beam will sustain such a load without injury during an indefinitely long period of time.

§ 3.—Let us now take a flexible chain consisting of an extremely large number of links; and, to simplify the reasoning, let us suppose that its weight is insignificantly small, so that this chain can be extended horizontally between two absolutely fixed points, *i.e.*, which cannot approach one another, whatever forces may act on them. If we now load the chain and, progressively increasing the vertical load, we observe the deflections, it will appear that there no longer exists the former direct relation between the load and the strain, but that the deflection varies as the cube root of the load; thus, on increasing the load eight times, the deflection will increase only twice. It is shown in statics that the tension of the material of a chain increases proportionally to the two-thirds power of the load, *i.e.*, if the load be increased eight times, the tension increases four times.

This relation between the deflections and the load will hold good so long as Hook's law for the material of the chain holds good; on a further increase of the load the deflections will increase more rapidly than before, and the tensions more slowly; these departures from uniformity are very considerable, so that the formulæ deduced for an absolutely elastic material cannot give either the magnitude of the deflection obtained or the tension just before the breaking of the chain.

§ 4.—In the Appendix are given two formulæ which determine the stresses in the chain corresponding to the different loads, if the material be regarded as absolutely elastic (formula 3a) and for real material (formula 5).

In Table A (see next page) the tensions q of a chain, whose sectional area is one square inch, and span $2\,a = 100$ in., corresponding to the various loads P, are shown (see Plate I. Fig. 1).

This example shows clearly what a great difference exists between the results given by the usual formulæ (3a) and (4) and the formulæ (5) and (6); the following statements are important :—

(1) When a horizontally stretched chain is loaded—the strains and stresses produced in it are not proportional to the load, but they increase more slowly than the load; the connection between these three quantities is defined by the two simple relations of statics (formulæ (3a) and (4)).

(2) When the tension of the chain has exceeded the limit of proportionality, the deflections increase more quickly and the tensions more slowly than is given by the preceding formulæ.

Thus, to the case of a stretched and loaded chain, our usual engineering criterion—the factor of safety—is *perfectly inapplicable*. If a load of 6 tons breaks our chain, the tension under a load of one ton will not be six times less than the ultimate tensile

TABLE A.

Tension q per Sq. In.	Absolutely Elastic Material.*		Real Material.†		
	Total Load P.	Maximum Deflection.	Relative Elongation.	Total Load P_1.	Maximum Deflection.
Lbs.	Tons.	In.	Percentage.	Tons.	In.
5,000	0·13	0·75	0·01	0·11	0·61
10,000	0·38	1·06	0·02	0·31	0·86
15,000	0·70	1·30	0·04	0·66	1·22
20,000	1·07	1·50	0·06	1·07	1·50
25,000	1·50	1·68	0·075	1·50	1·68
30,000	1·97	1·84	0·295	3·56	3·3
35,000	2·48	1·98	1·495	9·36	7·5
40,000	3·03	2·12	2·155	12·9	9·0
50,000	4·23	2·37	4·05	22·0	12·3
60,000	5·56	2·60	8·50	38·2	17·8
64,200	6·16	2·69	19·50	62·0	27·0

strength, but only $1\frac{3}{4}$ times; thus it considerably passes the limit of elasticity, and by repeated or indefinitely continued application of loads of one ton we may break our chain, supporting under ordinary conditions as much as 6 tons.

This circumstance presents a rare exception in engineering practice, and, in examining it, it is necessary to abandon the customary point of view, in order not to be misled.

§ 5.—Let us now return to the question of the flexion of thin plates supported at the edges by a rectangular frame.

If the plate rests freely on the frame and is not fastened to it, or if the frame is so weak that it may give way with the bending of the plate, the plate bends like a stiff

* The modulus of elasticity being equal to 33,000,000 lbs. per square inch.

† The elongations relate to a specimen of steel which has been actually tested at Watertown Arsenal. See Report of the Tests of Metals for Industrial Purposes. Washington, 1896, p. 153, No. 5,334.

beam, and there exists a direct proportion between the load and the tension at each point, almost up to the breaking of the plate, the deflections being supposed small when compared with the length and breadth of the plate. This case is the most disadvantageous and is hardly to be met with in shipbuilding practice.

If, however, the plate is firmly riveted to the frame, and the latter is so stiff as not to give way with the bending of the plate, we obtain the most interesting and practically important case of flexion. In this case the plate presents something intermediate between the limiting cases considered above of the stiff beam and the flexible chain, and, so long as the tensions are small, it very closely resembles the stiff beam; on increasing the load, the plate loses the property of bending like a beam, and the chief part of the resistance of the plate becomes due to its extension, like the case of the chain considered above.

It is the ratio of the least dimension of the plate to its thickness which determines whether the first or the second case gives a closer approximation; thus, when this ratio is 20 or less (for Siemens-Martin's steel) we have the first case, and when this ratio approaches 1,000, we have the case of the chain; for plates of medium thickness (when the said ratio varies from 50 to 200) their flexion is markedly different from both the extreme cases, as appears from Table B (see next page), which is calculated for a plate of Martin steel, the edges being so fastened to the supporting frame that they cannot shift horizontally, but the edge elements of the plate are free to incline. The thickness of the plate $t = \frac{3}{8}$ in., the breadth $2\,a = 48$ in., and its length is supposed very large. (See the formulæ 7*b*, 8*b*, and 9*a*, in the Appendix and Plate I. Fig. 2.)

This Table shows clearly how erroneous are the results given by both hypotheses (for example, 71 and 7·42 tons per square inch instead of 13·29 tons).

For pressures above 32 ft. of water column, the limit of elasticity is passed, the formula (9*a*) is no longer applicable; and it gives tensions which, as in the case of the chain, are greater than the true ones.

§ 6.—In the cellular-bottom construction, the plating may be considered as consisting of rectangular parts of plates, which are not simply supported on the edges, but clamped or built in, because the elements of the plate riveted to the frames are constrained to retain their original direction. In this case, we can observe the following curious circumstance :—

The greatest tension is here obtained, not in the middle of the plate, but at the edges, exactly as in a beam with fixed ends, which breaks under a uniform load at the ends, and not in the middle. Thus, under a load for which the tension at the edges approaches to the limit of elasticity, the tension at the middle of the plate, being almost one-half of the greatest, will be still far from it. On the load being increased, the tension at the edges will pass the limit of elasticity, and the material will approach the yielding point, *i.e.*, when the constant load causes an increasing elongation of the metal. Then all the boundary sections of the plate which hitherto retained their direction begin to incline, and thus the plate passes gradually to

TABLE B.

Pressure of Water Column.	Absolutely stiff Plate.	Real Plate.			Absolutely flexible Plate.	Absolutely stiff Plate.	Real Plate.	Absolutely flexible Plate.
	Stress due to Bending.	Stress due to Bending.	Stress due to Stretching.	Total Stress.	Stress due to Stretching.	Maximum Deflection.		
Height in Feet.	Tons per Square Inch.					Inches.		
½	1·11	0·97	0·11	1·08	0·46	0·10	0·08	0·16
1	2·22	1·56	0·32	1·88	0·74	0·19	0·14	0·21
2	4·43	2·33	0·69	3·02	1·17	0·38	0·21	0·26
4	8·87	3·18	1·34	4·52	1·86	0·76	0·29	0·33
8	17·74	4·14	2·41	6·55	2·95	1·53	0·39	0·41
12	26·6	4·74	3·32	8·06	3·86	2·29	0·45	0·47
16	35·5	5·33	4·12	9·35	4·68	3·06	0·51	0·52
20	44·3	5·61	4·86	10·47	5·42	3·82	0·55	0·56
24	53·2	5·88	5·57	11·45	6·12	4·58	0·58	0·60
28	62·0	6·17	6·23	12·40	6·79	5·35	0·61	0·63
32	71·0	6·44	6·85	13·29	7·42	6·11	0·64	0·66

the condition of a supported one, instead of a built-in one, as it was before. The tension in the sections which have given way immediately decreases, and the greatest tension passes to the middle of the plate.

After the removal of the load the plate will no longer return to its original position, and the permanent set in it at the edges will remain for ever. If the load be applied from the opposite side the plate changes its curvature for the opposite one, and at the same time a characteristic sound is heard, as may be observed when stepping on a thin warped plate.

As stated above, for a further increase of the load in the original direction, we shall have the flexion of the plate approaching to the case of a supported one, and we may calculate the greatest tension at the middle point of the plate by the corresponding formulæ, until this tension exceeds the limit of elasticity. On the load being still further increased, the tension thus calculated will be greater than the true one, and the plate before breaking will sustain a considerably greater pressure than that which would be given by these formulæ; the fracture always takes place near the middle of the plate.

§ 7.—For the rectangular plate built in on the boundary, even if it is regarded as absolutely stiff, the finding of the conditions of the flexion requires the integration of a partial differential equation of the fourth order, which has not yet been performed.* The investigation of the case of a real plate is, of course, yet more complicated, and I am compelled to assume the following postulate.

If we have four plates of the same thickness, having the following form of boundaries:—

(1) A rectangle with one side $2a$, the other being very long;

(2) A rectangle with one side $2a$, the other $2b$ $(b > a)$;

(3) An ellipse with the axes $2a$ and $2b$ $(b > a)$;

(4) A circle with the diameter $2a$;

all subjected to the same pressure, the corresponding stresses and strains in the first plate are greater than in second, in the second greater than in the third, and in the fourth they are the least. (See Plate I. Fig. 3.)

This postulate is justified for all the cases of flexion of thin plates already solved in the theory of elasticity; therefore, its extension to questions not yet theoretically solved will not be too hazardous a step. At any rate, this postulate admits of an easily realised experimental verification. (See Plate I. Fig. 4.)

This postulate being admitted, we may seek the limits of tension in a rectangular plate. By our formulæ, given in the Appendix, we can obtain the limiting value of the pressure which produces a permanent set of the plate and deprives it of its original elasticity, but the exact value of the breaking pressure cannot be calculated. In Table C (see next page) a limiting value of pressure admitted for all practical purposes is given; the formulæ which serve for the calculation of the Table and the assumptions used are explained in the Appendix. The greatest tension, $\sigma_{max.}$, is supposed to be $7\frac{1}{2}$ tons per square inch (1,200 kilog. per square cm.).

* Professor Koyalovich (of St. Petersburg University) has integrated this equation by means of Fourier's Series, whose convergency leaves much to be desired for practical calculation.

According to our postulate, the limiting pressure for a rectangular plate, whose breadth is $2a$ and length is $4a$ or more, is contained between the figures given in the second and third columns of this Table; these values are not very different from

TABLE C.

Ratio of Span to Thickness, $\frac{2a}{t}$	Rectangle, with the Sides $2a$ and ∞	Ellipse, with the Axes $2a$ and $4a$.	Circle of Diameter, $2a$.
	Height of Water Column in Feet.		
		(Nearly)	
60	23·5	28·9	62·5
70	17·1	21·0	45·8
80	13·1	16·1	34·8
90	10·4	12·8	27·6
100	8·4	10·4	22·4
110	7·0	8·6	18·6
120	6·0	7·4	15·8
130	5·2	6·4	13·6
140	4·6	5·6	11·8
150	4·1	4·9	10·5
160	3·7	4·4	9·4
170	3·3	4·0	8·5
180	3·0	3·6	7·7

one another, the maximum difference being only 20 to 23 per cent., and for the riveted plates, neglecting the weakening of the section by the rivet holes (which gives 12 to 22 per cent.), we may adopt the first limit for all rectangular plates, in which the ratio of the length to the breadth is 2 or more.*

* Dr. F. Grashof ("Theorie der Elasticität und Festigkeit," Berlin, 1878, p. 369) gives, for a stiff rectangular plate with sides $2a$ and $2b$ ($b > a$) and thickness t, the formula,

$$\sigma_{\text{max.}} = 2p\left(\frac{a}{t}\right)^2 \frac{1}{1+\left(\frac{a}{b}\right)^4}$$

where $\sigma_{\text{max.}}$ is the maximum tension, p the pressure on unit of area. If we assume here $b = \infty$ and $b = 2a$, the tension $\sigma_{\text{max.}}$ in these two cases will differ by only 6 per cent. The truth of this formula is doubtful, and I quote it not as a proof, but by way of reference.

§ 8.—Thus, with a thickness of the outer shell plating of a large battleship where $t = \frac{16}{20}$ in., and with a space between the transversal frames $2a = 48$ in., the admissible pressure is approximately equal to 24 ft. of salt water, and is almost independent of the space between the longitudinal frames, if this space is 8 ft. or more. Therefore, in still water, the draught of the ship being 25 ft. to 26 ft., the pressure approaches to the limiting one, whilst the maximum tension amounts to about 8 tons per square inch. For smaller ships, the shell plating of which has a thickness of $\frac{10}{20}$ in. to $\frac{12}{20}$ in., the space between the transverse frames being the same, and the draught of water 20 ft. to 25 ft., the tension exceeds the above limit, and attains, in some cases, 10 tons to 12 tons per square inch and more.

The inner bottom plates of all ships have a thickness of $\frac{6}{20}$ in. to $\frac{8}{20}$ in., with the same frame spaces, and the admissible pressure is from 4 ft. to 6 ft. of water column; but, as by some requirements they are tested by a pressure of nearly 20 ft. to 25 ft., a permanent set of these plates at the frames is, of course, inevitable, and every engineer who has examined the inner bottom plates, after a trial by the pressure of such a height of water column, knows this. The tension exceeds the limit of elasticity, and attains 12 tons to 15 tons per square inch and more.

For the mercantile marine, Lloyd's rules require very small spaces between the girders, and also intermediate transverse frames. This requirement decreases the tension on the plating by about twice or three times.

§ 9.—In calculating the tension of the bottom and deck plating, produced by the bending moments due to inequalities of weight and buoyancy, it is assumed that all the longitudinal plates take a whole part in the resistance of the ship considered as a solid beam, a maximum stress of 4 tons to 5 tons, or even 6 tons per square inch being usually admitted. It must, however, be remembered that to this stress must be added the preliminary one, proceeding from the pressure of the water. Thus the total stress of the bottom plating may be much nearer to 12 tons or 15 tons per square inch than the required 5 tons.

Such a stress is scarcely desirable, if we bear in mind that one-half of the stress in a pitching ship is a rapidly varying one, not only in magnitude but also in direction (tension and compression alternately). In civil engineering constructions, such an intensity of tress is never allowed.

If we wish to reduce these values, it ought to be remarked that, the stresses caused by pitching being inevitable, it is possible to reduce only those which are due to the hydrostatic pressure. Accordingly, I venture to propose a practical solution of the following problem :—

To construct the hull of a ship so that the tension on the inner bottom plates, when exposed to the pressure of the test column of water,* should not exceed the practical admitted limits, and that the tension on the outer shell plating due to the same hydrostatic pressure of water should be as small as desired.

It is evident from our formulæ that, in order to diminish the tension in the plating without increasing its thickness, it is necessary either to reduce the span between the transverse or longitudinal frames, or to increase the stiffness of the plates, by riveting to them some additional ribs. A reduction of the span would make the cellular-bottom system unrealisable, it is the second method therefore which remains. In order to preserve the total weight of ribbed plating the same as before, the thickness of the plates is to be duly reduced and the ribs may be placed longitudinally and made continuous, even under all the transverse frames, except the watertight ones.

Fig. 6 (Plate II.) shows a scheme of such a system of construction.

§ 10.—The floors in this system may be either lightened by cutting holes, or constructed upon the bracket system, but the outer angle-bars and the reverse frames must be made continuous. This considerably simplifies the fitting together and the verification of the frames. These angle-bars are bevelled as usual, and on the outer and inner flanges of the frames additional longitudinal ribs are laid continuously; to these ribs the plates of the outside and inner bottom plating are riveted. In order to fasten the transverse frames to the plating, short unbevelled pieces of the same section as the ribs are riveted to the angle-bars of the frames.

The additional ribs are to be calculated by the ordinary formula—

$$W = \frac{2}{3}\frac{p}{\sigma}a\,b^2,$$

where W is the moment of resistance of the profile, $2a$ and $2b$ are the spaces between the ribs and the transverse frames respectively, p the water pressure, and σ the admitted tension. This formula relates to a beam with two fixed ends loaded uniformly, the load being the pressure of water on the corresponding strip of plate. The thickness t of the plating is to be calculated by the formula

$$t = a\sqrt{1{\cdot}82\,\frac{p}{\sigma}} = \frac{2a}{105}\sqrt{\frac{H}{\sigma}}, \qquad (32)$$

where H is the height of water column in feet, and σ is the tension in tons per square inch. This formula is explained in the Appendix.

* The height of this column being equal to the draught of the vessel, or more.

In the ordinary system the longitudinal frames, besides their immediate destination—the connection of the transverse frames—have a great importance in another respect. We have shown in the Appendix how greatly the tension in a plate subjected to the pressure of water is diminished by reason of the stays between the supports; it is therefore very important to prevent any bringing nearer together of the transverse frames and any tendency of the ship to hog, which would arise therefrom; but when the plating is sufficiently stiff by itself and the bending from the water pressure is insignificant, there is no need for continuous longitudinal frames, sufficient mutual connection of the transverse frames being obtained by means of longitudinal bulkheads, the vertical keel plate, the watertight continuous stringer, and the intercostal longitudinal stringers. These latter may even be replaced by diagonal stays riveted to the ribs in such a manner that in places where a great weight is concentrated on several frames (for example, the side turrets) these stays may be fitted closer.

As to the weight of the ribbed plating proposed, it does not exceed that of the usual system,* the space between the added ribs being sufficiently small. The following Table D gives the resulting weights, the space between the transverse frames being supposed to be 4 ft.

TABLE D.

Space between the added Ribs.	Outer Shell Plating.				Inner Bottom Plating.				Total Weight, per Sq. Ft.
	Weight of Plates, per Sq. Ft.	Z-bar added Ribs.*		Total Weight, per Sq. Ft.	Weight of Plates, per Sq. Ft.	Z-bar added Ribs.*		Total Weight, per Sq. Ft.	
		Height.	Cross-Section.			Height.	Cross-Section.		
In.	Lbs.	In.	Sq. In.	Lbs.	Lbs.	In.	Sq. In.	Lbs.	Lbs.
12	15	5	2·5	23½	9	3	1·4	14	37½
16	20	5½	3·1	28	12	3½	1·6	16	44
20	25	6	3·5	32	15	4	1·7	18½	50½
24	30	6½	3·9	36½	18	4	2·1	21½	58

* The stiffness of Z-bars is increased by about 25 per cent., due to the riveted plate.

The maximum tension proceeding from the pressure of a water column 26 ft. in height is 2½ tons per square inch for the outer shell plating, and 7½ tons per square inch for the inner bottom plating. It must be remarked that here the tension in the shell plating is directed across the ship, and is to be combined with the longitudinal tension, not by simple addition, but by the parallelogram law; thus, if the longitudinal

* The weight of both platings with longitudinal stringers (girders) for a big ship constructed upon the usual system is about 45 lbs. to 50 lbs. per square foot and more.

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tension arising from pitching is 5 tons per square inch, the maximum tension will be $\sqrt{5^2 + (2\frac{1}{2})^2}$, or close upon 5·6 tons per square inch, and not $7\frac{1}{2}$ tons per square inch.

§ 11.—The plating in the proposed system (including the additional ribs) is considerably stronger than usual, because their flexural rigidity (for bending) increases in the ratio of the height of the strengthening ribs to the thickness of the plates (*i.e.*, 7 to 10 times). Such an increase of strength is important when the ship comes in contact with the ground in shallow places.

The thickness of the plates themselves is reduced almost to one half, but I do not think that such a reduction could call forth serious objections, because there does not exist a law that big ships should have thicker plating, if this is not required by the conditions of strength. There have been small ships with thin plating, doing excellent service for dozens of years, being carefully preserved from corrosion below the water line. Moreover, the rusting of plates, when they are thin, does not produce a worse effect by increasing the tension, as in the case of thick ones; thus, for example, if in a $\frac{16}{20}$ in. plate, $\frac{2}{20}$ in. be destroyed by rust, the tension will increase in the ratio of $(\frac{16}{14})^2$, that is, it will be 9·1, instead of 7 tons per square inch; in a thin $\frac{8}{20}$ in. plate, on the other hand, subjected to a tension of only $2\frac{1}{2}$ tons per square inch, the reduction of the thickness by $\frac{2}{20}$ in. will bring the tension to $2\frac{1}{2} \times (\frac{8}{6})^2$, or 4·4 tons per square inch. I do not think the latter can be considered worse than the former.

In the existing system, in places where the transverse and longitudinal frames are secured, there is formed a very rigid connection between the bottoms, which deprives the inner bottom of any importance, because even by a slight blow against a rock both bottoms are then broken through together. In the proposed system, where the longitudinal frames are replaced by diagonal stays, the transverse frames are only strong enough to withstand all the forces acting upon the ship under the ordinary conditions of her service, and thus these frames may be broken by the blow, whilst the inner bottom will still remain intact.

In describing this system of construction, I do not consider it exempt from many imperfections, and I wish only to show one of the practically possible solutions of the problem stated above, without any increase of weight or difficulty of work. A perfect system of construction can only be elaborated by the united efforts of many practical shipbuilders during its actual carrying out.

APPENDIX.

THE elementary theory of thin plates solves the question of their flexion with the following assumptions:—

(1) The middle surface is unextended.

(2) The line elements of the plate initially normal to the middle surface remain straight and normal to this surface after strain.

(3) There is no normal traction across the planes parallel to the middle surface.

I will attempt to do without the first assumption.

I.—THE RECTANGULAR PLATE.

Let us take a thin plate having a thickness t, loaded with a uniform normal pressure p on a unit of its area; let its boundary be a rectangle, one side of which is equal to $2a$, and the other very great in comparison with the first. Then the middle part of this plate, being sufficiently removed from its short side, will, after flexion, present the surface of a cylinder, and, if we separate a narrow strip by two planes perpendicular to the faces of the plate, and parallel to the short sides of the rectangle, the conditions of its flexion will be just the same as for the strips bounding it on both sides. We will at first regard its flexion independently of the remaining part of the plate, and let us take the breadth of this strip equal to a unit of length.

If the two long sides of the rectangle are placed under such conditions that there are certain obstacles to their approaching one another, the reaction at each point of the boundary will be directed not parallel to the external pressures acting upon the plate, but at an angle to them, the direction of reaction being inclined outwards from the middle of the plate. We will resolve it into two components, one parallel to the external pressure, equal to $-pa$ (the breadth of the strip being equal to unity); the other perpendicular to the long side of the plate, equal to qt, where q is the unknown tension on a unit of area of the transverse section of the strip, assumed to be uniformly distributed over this section.

Directing the x axis perpendicular to the long side of the plate in the middle surface, and the y axis coinciding to the reaction pa, we will denote by—

M, the bending moment in the section whose abscissa is x;

ϕ, the angle of inclination of this section after flexion, reckoned from the initial direction of the section considered;

M_0 and ϕ_0, the same quantities for the section passing through the origin of co-ordinates;

I, the moment of inertia, and F, the area of the cross-section of the strip;

E, the modulus of elasticity (Young's modulus); the quantities M, M_0, I, and F referring to a strip of unit breadth.

Then, observing that the deflections y at each point are extremely small compared with the span $2\,a$, we have—

$$\frac{d\,\phi}{d\,x} = \frac{M}{E\,I} = \frac{1}{E\,I}\left(M_0 - p\,a\,x + \frac{p\,x^2}{2} + q\,F\,y\right);$$

but, as—

$$\frac{d\,y}{d\,x} = \tan\phi = \phi \text{ nearly,}$$

then, differentiating our equation in regard to x, we obtain—

$$\frac{d^2\,\phi}{d\,x^2} = \frac{1}{E\,I}\{-p\,a + p\,x + q\,F\,\phi\},$$

and consequently, calling, for simplicity,

$$\frac{q\,F\,a^2}{E\,I} = u, \tag{A}$$

we find—

$$\phi = A\,\text{Ch}\,\sqrt{u}\frac{x}{a} + B\,\text{Sh}\,\sqrt{u}\frac{x}{a} + \frac{p\,a^2}{E\,I\,u}(a - x),$$

where—

$$\text{Ch}\,k = \frac{e^k + e^{-k}}{2}, \quad \text{Sh}\,k = \frac{e^k - e^{-k}}{2},$$

while A and B are arbitrary constants, determined from the conditions—

$$\left(\frac{d\,\phi}{d\,x}\right)_{x=0} = \frac{M_0}{E\,I} = B\,\frac{\sqrt{u}}{a} - \frac{p\,a^2}{E\,I\,u}$$

$$\left(\phi\right)_{x=a} = 0 = A\,\text{Ch}\,\sqrt{u} + B\,\text{Sh}\,\sqrt{u},$$

or, making the necessary substitutions—

$$\phi = -\frac{M_0\,u + p\,a^2}{q\,F\,a}\cdot\frac{\text{Sh}\,\sqrt{u}\left(1 - \frac{x}{a}\right)}{\sqrt{u}\,\text{Ch}\,\sqrt{u}} + \frac{p\,(a - x)}{q\,F}, \tag{1}$$

and

$$\phi_0 = -\frac{M_0\,u + p\,a^2}{q\,F\,a}\,\frac{\text{Sh}\,\sqrt{u}}{\text{Ch}\,\sqrt{u}} + \frac{p\,a}{q\,F}. \tag{1A}$$

The unknown quantity q entering into this expression will be determined according to the displacement of the supports along the x axis; calling the displacement along this axis of a point whose abscissa is x, $\delta\,x$, the displacement of the support, $\delta\,x_0$, we have—

$$\frac{d\,\delta\,x}{d\,x} = \left(1 + \frac{q}{E}\right)\cos\phi - 1 = -\frac{1}{2}\phi^2 + \frac{q}{E},$$

discarding higher powers of ϕ.

Observing that from symmetry

$$(\delta\,x)_{x=a} = 0,$$

and integrating the preceding expression within the limits 0 and a, we find—

$$\delta x_0 = -\frac{q a}{E} + \frac{p^2 a^5}{6 q^2 F^2} + \left(\frac{M_0 u + p a^2}{q F \sqrt{a}}\right)^2 \left\{\frac{\text{Th} \sqrt{u}}{4 u \sqrt{u}} - \frac{1}{u (2 \text{Ch} \sqrt{u})^2}\right\}$$

$$-\frac{p a}{q F}\left(\frac{M_0 u + p a^2}{q F}\right) \frac{\sqrt{u} - \text{Th} \sqrt{u}}{u \sqrt{u}}, \qquad (2)$$

where—

$$\text{Th}\, k = \frac{\text{Sh}\, k}{\text{Ch}\, k} = \frac{e^k - e^{-k}}{e^k + e^{-k}}.$$

The equations (1A) and (2) completely solve the question of the flexion of our strip, if the mode of securing its ends to the support is indicated. Geometrically, this mode of securing is expressed by the quantities ϕ_0 and δx_0; from the point of view of statics—by the quantities M_0 and q; when two of these quantities are given, the two remaining can be determined by these equations.

Let us examine certain limiting cases:—

(1) Absolutely Stiff Plate.

$$q = 0.$$

If the horizontal re-action $q = 0$, *i.e.*, the supports may approach when the plate is bent, and in virtue of the relation (a), the "argument" u is also equal to zero; therefore, in order to avoid the indetermination, we must expand in the equation (1) the exponential functions—

$$\text{Sh} \sqrt{u}\left(1 - \frac{x}{a}\right) \text{ and } \text{Ch} \sqrt{u}$$

in series, proceeding in ascending powers of u, and then we shall obtain—

$$\phi = \frac{a - x}{6 E I}\left\{p (2 a^2 + 2 a x - x^2) - 6 M_0\right\}.$$

Assuming in this expression $M_0 = 0$, we have

$$\phi = \frac{p}{6 E I}(a - x)(2 a^2 + 2 a x - x^2);$$

but if we assume that $M_0 = \frac{p a^2}{3}$, which correspond to $\phi_0 = 0$, we shall find—

$$\phi = \frac{p}{6 E I} x (a - x)(2 a - x).$$

The last two formulæ completely coincide with the corresponding formulæ for the flexion of stiff beams; the first corresponds to the case of supported ends, and the second to that of fixed ends.

The displacement of the supports δx_0, may be found by the equation (2), but it is of little interest to us.

(2) Absolutely Flexible Plate.

$$u = \infty. \qquad \partial x_0 = 0.$$

Let us suppose—

$$\frac{q \, F \, a^2}{E \, I} = u = \infty \; ;$$

as no factor of the numerator can be infinite, the equation written is only possible when

$$I = 0,$$

i.e., when our plate is absolutely flexible, and consequently the bending moment M in every section of it will be zero; assuming in the equation (2) $M_0 = 0$ and $u = \infty$, we obtain—

$$q = \sqrt[3]{\frac{1}{6} p^2 E \left(\frac{a}{F}\right)^2} \qquad (3)$$

and consequently for the plate, where $F = t$—

$$q = \sqrt[3]{\frac{1}{6} p^2 E \left(\frac{a}{t}\right)^2} \qquad (3\text{A})$$

The equation (1), after a similar substitution and integration between 0 and x, will become

$$y = \frac{p}{2 \, q \, t} \, x \, (2 \, a - x) = \sqrt[3]{\frac{3}{4} \frac{p}{E} \left(\frac{t}{a}\right)^2} \, \frac{x \, (2 \, a - x)}{t} \qquad (4)$$

This equation represents a parabola of the second order; the pressure of the water on the plate is always normal to its surface, and the corresponding catenary must be an arc of a circle; the parabola is obtained instead of the circle, because we have considered a vertical pressure instead of a normal, but the difference between the arc of the circle and the obtained arc of the parabola is insignificantly small, and, within practical limits, has no influence on the results.

It is easy to show that these equations coincide with the ordinary equations of a catenary,

$$d \left(q \, F \frac{dx}{ds}\right) = 0,$$

$$d \left(q \, F \frac{dy}{ds}\right) = p \, ds,$$

where $d \, s$ is an element of arc, if we assume, observing that the height of the catenary is extremely small compared with its span,

$$ds = dx,$$

and determine the tension q from the equation, analogous to equation (2)—

$$\frac{a \, q}{E} = \frac{1}{2} \int_0^a \left(\frac{dy}{dx}\right)^2 d \, x.$$

The following observation is also of interest; it is not difficult to see that our catenary is extremely near to a circle with a radius—

$$R = \frac{q}{p} t,$$

which gives—

$$t = R\frac{p}{q}.$$

This is Rankine's formula for thin cylindrical shells subjected to internal pressure.

We have deduced all these formulæ on the ordinary assumption of the theory of elasticity, *i.e.*, the proportionality of the forces of elasticity and the deformations caused by them. Availing ourselves of the fact that in the last case the tensions at all the points of the plate are the same, we may solve the question also without this assumption. And, as the formulæ (3) and (4) are statical formulæ, the letter E in them must be regarded as the ratio of the tension on a unit of area q to the corresponding relative elongation, which we will call ϵ, independently of the elastic properties of the material.

Therefore, making the above-mentioned substitution, we shall obtain—

$$p = q\frac{t}{a}\sqrt{6\,\epsilon} \qquad (5)$$

$$y_{max.} = \frac{p\,a^2}{2\,q\,t} = a\sqrt{\frac{3}{2}\epsilon} \qquad (6)$$

The magnitude of the relative elongation ϵ may be obtained by testing the material on an ordinary testing machine.

Let us also observe that these formulæ may be applied as long as the deflection $y_{max.}$ is small in comparison with the span $2\,a$, because otherwise the tension q will not be the same throughout the whole length of the chain, and then it is necessary to apply to the catenary the more complex formulæ given in text-books on statics. If

$$\frac{y_{max.}}{2\,a} = \frac{1}{8},$$

the maximum tension q_0, at the ends of the chain will be approximately 10 per cent. less than q_a in the middle. However, for our question, these formulæ are of no importance, as in the bent plates the deflections even in case of rupture do not attain such considerable magnitudes.

(3) Real Plate.

$$M_0 = 0. \qquad \delta\,x_0 = 0.$$

This case corresponds to the frames which cannot absolutely approach each other, but which at the same time allow the long edge elements of the plate to incline, as on ideal hinges, without friction.

Then, the equation (2), with the help of the relation (A), will give, after certain transformations—

$$\frac{17}{630}\left(\frac{p}{E}\right)^2\frac{a^8\,F}{I^3} = \frac{\frac{17}{630}u^4}{\frac{5}{4}\left(\frac{\mathrm{Th}\,\sqrt{u}}{\sqrt{u}} - 1\right) + \left(\frac{\mathrm{Th}\,\sqrt{u}}{2}\right)^2 + \frac{1}{6}u} = F_1(u). \qquad (7)$$

On the first side of this equation are the quantities known to us, *i.e.*, those depending upon the dimensions of the plate (a, F, I), its material (E), and load (p), while on the second side those depending only on the "argument" u; consequently this equation may be used for the determination of this last. Unfortunately, the transcendental form of this function $F_1(u)$, does not allow of a general expression being given for u in terms of known quantities; and, in order to solve this equation for various values of the "argument" u from 0 to 20, the Table following of the values of $F_1(u)$ has been calculated. This Table makes it possible by interpolating by second differences to find the value of the "argument" u for a given value of $F_1(u)$ correct to $\frac{1}{2}$ per cent. For the value of the "argument" u greater than 20, the equation is solved by the usual method of successive approximations, assuming the first approximation—

$$(\text{Th}\sqrt{u})_{u>20} = 1.$$

The "argument" u being known, we find the longitudinal tensions q of the plate from the relation—

$$q = \frac{\text{E I}}{\text{F}\,a^2}u. \qquad (\text{A})$$

Integrating now the equation (1) between the limits 0 and x, we obtain the equation of the line of elasticity—

$$y = \frac{p\,a^4}{\text{E I}}\left\{\frac{\text{Ch}\sqrt{u}\left(1-\frac{x}{a}\right)-\text{Ch}\sqrt{u}}{u^2\,\text{Ch}\sqrt{u}} + \frac{x\,(2\,a-x)}{2\,a^2\,u}\right\}^*$$

and the maximum deflection, corresponding to $x = a$—

$$y_{\text{max.}} = \frac{p\,a^4}{\text{E I}}\left\{\frac{1}{2\,u} - \frac{1}{u^2} + \frac{1}{u^2\,\text{Ch}\sqrt{u}}\right\} \qquad (8)$$

In the same manner we shall obtain the maximum angle of inclination ϕ_0 putting $x = 0$—

$$\phi_0 = \frac{p\,a^3}{\text{E I}}\left(1 - \frac{\text{Th}\sqrt{u}}{\sqrt{u}}\right),$$

and the maximum bending moment at $x = a$ —

$$\text{M}_{\text{max.}} = p\,a^2\left(\frac{1}{u} - \frac{1}{u\,\text{Ch}\sqrt{u}}\right).$$

The last four equations, when $u = 0$ (stiff plate), and after removing the ambiguities of the form $\infty - \infty$, will give the corresponding equation for a stiff beam with uniform load and supported by the two ends.

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* This formula, and also (9A), (11A), and (12A), completely coincide with the formulæ given by Professor G. H. Bryan (see Transactions of the I.N.A., 1891, pp. 206 and 207), but the method of calculation of the quantities q and u is not shown there.

If we investigate the case of a thin plate of a thickness t without the strengthening ribs, we must assume

$$F = t \text{ and } I = \frac{1}{12} t^3,$$

and thus our formulæ are altered as follows:—

$$q = \frac{E u}{12} \left(\frac{t}{a}\right)^2 \qquad (A_1)$$

$$\frac{1632}{35} \left(\frac{p}{E}\right)^2 \left(\frac{a}{t}\right)^8 = F_1 (u) \qquad (7A)$$

$$y_{max.} = \frac{5 p}{2 E} \left(\frac{a}{t}\right)^4 t \left\{ \frac{24}{5} \left(\frac{1}{2u} - \frac{1}{u^2} + \frac{1}{u^2 \operatorname{Ch} \sqrt{u}}\right) \right\} = \frac{5 p}{2 E} \left(\frac{a}{t}\right)^4 t \, \phi_1(u), \qquad (8A)$$

where the factor $\phi_1 (u)$ depends only upon the "argument" u given below in the Table.

The maximum tension $\sigma_{max.}$ will be determined by the relation

$$\sigma_{max.} = q + \frac{6 M_{max.}}{t^2}$$

or, after substitutions and simplifications,

$$\sigma_{max.} = 3 p \left(\frac{a}{t}\right)^2 \left\{ \frac{1}{9} \sqrt{\frac{102}{35}} \frac{u}{\sqrt{F_1(u)}} + \frac{2}{u} \left(1 - \frac{1}{\operatorname{Ch} \sqrt{u}}\right) \right\} = 3 p \left(\frac{a}{t}\right)^2 f_1 (u). \qquad (9)$$

The factor $f_1 (u)$ is calculated for the same values of u from 0 to 20, and is given in the same Table.

We note that

$$\varphi_1 (0) = 1, \quad \text{and } f_1 (0) = 1 ;$$

thus, if in the formulæ (8A) and (9) we reject these factors, they will refer to stiff beams.

The equations (7A), (8A), and (9) determine the maximum deflection and tension in the case of the flexion of a thin plate placed in the conditions above stated; the tables of magnitudes

$$F_1(u), \ \phi_1(u), \text{ and } f_1(u)$$

simplify extremely the practical calculations, permitting the question to be solved as quickly as for the stiff beam.

(4) Real Plate.

$$\phi_0 = 0, \quad \text{and } \delta x_0 = 0.$$

This case corresponds to an absolutely stiff frame, the plate being clamped or built-in on the long edges. The requisite conditions may occur if the plate covers several cells in contact with their long sides.

The equation (1) when $x = 0$ gives

$$\phi_0 = -\frac{M_0 u + p a^2}{q F a}\frac{\mathrm{Th}\sqrt{u}}{\sqrt{u}} + \frac{p a}{q F} = 0,$$

whence we shall find M_0, and, substituting in equation (2), shall obtain as before

$$\frac{1}{945}\left(\frac{p}{E}\right)^2\frac{a^8 F}{I^3} = \frac{\frac{1}{945}u^4}{\left\{\frac{5}{12}u + \frac{25}{16} - \left(\frac{3}{4} + \frac{\sqrt{u}}{2\,\mathrm{Th}\sqrt{u}}\right)^2\right\}} = F_2(u), \tag{10}$$

an equation from which we determine u, if it is between the limits 0 and 6, by means of the tables; if, however, $u > 6$, by means of the method of consecutive approximations. Knowing this "argument," we find q from the relation (A), and then obtain as before—

$$y_{\max.} = \frac{p a^4}{E I}\frac{1}{u}\left\{1 - \frac{\mathrm{Th}\frac{\sqrt{u}}{2}}{\frac{\sqrt{u}}{2}}\right\}$$

Without dwelling any longer on the general case of the plate strengthened by ribs, and passing to the plate of a thickness t without strengthening, we shall obtain in an analogous manner—

$$\frac{64}{35}\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 = F_2(u). \tag{10A}$$

$$q = \frac{E u}{12}\left(\frac{t}{a}\right)^2. \tag{A$_1$}$$

$$y_{\max.} = \frac{1}{2}\frac{p}{E}\left(\frac{a}{t}\right)^4 t\left[\frac{12}{u}\left(1 - \frac{\mathrm{Th}\frac{\sqrt{u}}{2}}{\frac{\sqrt{u}}{2}}\right)\right] = \frac{1}{2}\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_2(u) \tag{11}$$

$$\sigma_{\max.} = 2p\left(\frac{a}{t}\right)^2\left\{\frac{1}{3\sqrt{35}}\frac{u}{\sqrt{F_2(u)}} + \frac{3}{u}\left(\frac{\sqrt{u}}{\mathrm{Th}\sqrt{u}} - 1\right)\right\} = 2p\left(\frac{a}{t}\right)^2 f_2(u). \tag{12}$$

The factors $\phi_2(u)$ and $f_2(u)$ are inserted in the Table below; while, as before—

$$\phi_2(0) = 1, \text{ and } f_2(0) = 1.$$

Let us also remark that in this case $\sigma_{\max.}$ is near to the fastened end of the plate, and it is not difficult to show that, as long as this tension is less than the limit of elasticity of the material and the condition $\phi_0 = 0$ holds good, the absolute value of the tension at this point is greater than (almost twice) its corresponding value at the middle of the plate. The yielding point of the material being passed, the circumstances of bending of the plate approach those of the previous case.

Putting in the equations (7A) and (10A) $u = \infty$ (case of a chain), we may obtain the same equation (3A), although they refer to various cases of the plate with supported and fixed edges. This is clear, because for an absolutely flexible plate there cannot be different methods of fastening the edges. In the same way the equations of the elastic line (or deflection) (8A) and (11) will coincide with the equation (4), as soon as we put in them u equal to infinity.

All our formulæ are deduced for a strip independent of the remaining parts of the plate, in order to show clearly their connection with the ordinary case of beam flexion. If we wish to pass to the case of a long plate our differential equation will take the following form—

$$\frac{d^2 \phi}{d x^2} = \frac{1 - \mu^2}{\mathrm{E\,I}} (- p\, a + p\, x + q\, \mathrm{F}\, \phi),$$

where μ is Poisson's ratio; and putting, for simplicity—

$$\frac{q\, a^2\, \mathrm{F}}{\mathrm{E\,I}} (1 - \mu^2) = u_1, \qquad (\mathrm{A}_2)$$

$$p\,(1 - \mu^2) = p_1,$$

we obtain the same equation as before. Thus our principal equations will be modified as follows:—

$$\frac{1632}{35} \left(\frac{p}{\mathrm{E}}\right)^2 \left(\frac{a}{t}\right)^8 (1 - \mu^2)^3 = \mathrm{F}_1 (u)\,; \qquad (7\mathrm{B})$$

$$q = \frac{\mathrm{E}\, u}{12} \left(\frac{t}{a}\right)^2 \frac{1}{1 - \mu^2}\,; \qquad (\mathrm{A}_2)$$

$$\sigma_{\mathrm{max.}} = 3 p \left(\frac{a}{t}\right)^2 (1 - \mu^2) f_1 (u)\,; \qquad (9\mathrm{A})$$

$$y_{\mathrm{max.}} = \frac{5}{2} \frac{p}{\mathrm{E}} \left(\frac{a}{t}\right)^4 t\,(1 - \mu^2)\, \phi_1 (u). \qquad (8\mathrm{B})$$

The equations (10A), (11), and (12) will be modified likewise.

II.—The Circular Plate.

Let us take a thin plate of a thickness t, bordered by a circular contour of radius a, and loaded with a uniform pressure on the unit of area p. If the plate is secured to the supporting frame, so that it cannot slide on it, and the frame itself is so stiff that it cannot be deformed, the reaction of the latter will be directed, not in the direction parallel to the external pressure acting upon our plate, but will be deflected from the centre of the plate.

Let us denote by q the component of this reaction directed along the radius of the circle and belonging to a unit of area of the section of the plate at the contour. We will take the middle plane of the plate for the plane $x\,z$, the centre of the circle lying in this plane as the origin of co-ordinates, the axes x and z being directed along two rectangular radii, and the y axis perpendicular to the middle plane.

The normal tension along the axis of x will be denoted by σ_x, and the relative elongation of the fibre in the same direction by ϵ_x; the tensions and elongations along the other axes will be denoted by the same letters with the corresponding indices. If further we denote by E the modulus of elasticity

(Young's modulus) and by μ Poisson's ratio, at each point of the plate, these quantities will be connected by the known relations—

$$\sigma_x = \frac{E}{1+\mu}\left(\epsilon_x + \frac{\mu\,\theta}{1-2\mu}\right);$$

$$\sigma_y = \frac{E}{1+\mu}\left(\epsilon_y + \frac{\mu\,\theta}{1-2\mu}\right);$$

$$\sigma_z = \frac{E}{1+\mu}\left(\epsilon_z + \frac{\mu\,\theta}{1-2\mu}\right),$$

where

$$\theta = \epsilon_x + \epsilon_y + \epsilon_z.$$

But, as the thickness of the plate t is assumed to be extremely small in comparison with its radius a, the tension σ_y will be small as compared with the quantities σ_x and σ_z. Therefore—

$$\left.\begin{aligned} \sigma_x &= \frac{E}{1-\mu^2}(\epsilon_x + \mu\,\epsilon_z) \\ \sigma_z &= \frac{E}{1-\mu^2}(\epsilon_z + \mu\,\epsilon_x) \end{aligned}\right\} \qquad (13)$$

It is easy to see that, after flexion, the middle plane of the plate will become a surface of revolution; if we denote by ρ the radius of curvature of its meridian, by ρ_1 the radius of curvature of this surface in a perpendicular direction, and if we distinguish by the index ^{0} the tensions and strains for the points of our middle-surface, we have—

$$\epsilon_x = \epsilon_x{}^0 + \frac{l}{\rho}; \qquad \epsilon_y = \epsilon_y{}^0 + \frac{l}{\rho},$$

where l is the distance of some point of the plate from the middle surface, while evidently

$$\frac{1}{2}t > l > -\frac{t}{2}.$$

But, as the deflections of the plate are extremely small compared with its radius—

$$\frac{1}{\rho} = -\frac{d^2 y}{d x^2}, \quad \frac{1}{\rho_1} = -\frac{1}{x}\frac{d y}{d x};$$

observing, further, that—

$$\epsilon_x{}^0 = \epsilon_y{}^0 = q$$

we shall obtain finally—

$$\left.\begin{aligned} \sigma_x &= q - \frac{E\,l}{1-\mu^2}\left(\frac{d^2 y}{d x^2} + \frac{\mu}{x}\cdot\frac{d y}{d x}\right), \\ \sigma_z &= q - \frac{E\,l}{1-\mu^2}\left(\frac{d y}{d x}\cdot\frac{1}{x} + \mu\frac{d^2 y}{d x^2}\right), \\ \epsilon_x &= \frac{q}{E}(1-\mu) - l\frac{d^2 y}{d x^2}, \\ \epsilon_y &= \frac{q}{E}(1-\mu) - \frac{l\,d y}{x\,d x}. \end{aligned}\right\} \qquad (14)$$

Let us now consider an elementary volume in our plate, bounded—

(1) By two infinitely near surfaces, parallel to the middle surface, and distant $d\,l$ from each other;

(2) By two radial surfaces forming an angle $d\,a$; and

(3) By two cylindrical surfaces, whose axis coincides with the axis of y, and the radii are x and $x + d\,x$.

Then, denoting by τ_z the shearing stress on a unit of area of the surface perpendicular to the axis y, and projected on the axis of x, we shall obtain the following equation of equilibrium for all the forces of elasticity acting on our elementary volume—

$$\frac{1}{x}\frac{d\,(x\,\sigma_x)}{d\,x} - \frac{\sigma_y}{x} + \frac{d\,\tau_z}{d\,l} = 0,$$

whence, integrating between the limits $\pm\,\frac{1}{2}\,t$, and, observing that $\tau_z = 0$ on the faces of the plate, we find, making the necessary substitution—

$$\tau_z = \frac{E}{1-\mu^2}\,\frac{t^2 - 4\,l^2}{8}\left(\frac{d^3 y}{d\,x^3} + \frac{1}{x}\frac{d^2 y}{d\,x^2} - \frac{1}{x^2}\frac{d\,y}{d\,x}\right) \qquad (15)$$

If we cut out of our plate a disc having a radius x, and reject the ring surrounding it, then the equality of the projections on the axis y, of the external faces acting on the disc on the one hand, and of the internal faces acting on the contour of the section on the other hand, will give us the following equation—

$$\int_{-\frac{t}{2}}^{+\frac{t}{2}} \tau_z\,.\,2\,\pi\,x\,.\,d\,l + \int_{-\frac{t}{2}}^{+\frac{t}{2}} \sigma_x\,\frac{dy}{dx}\,.\,2\,\pi\,x\,.\,d\,l = \pi\,x^2\,p.$$

After integrating and simplifying, we shall obtain the differential equation of the meridian—

$$\frac{d^3 y}{d\,x^3} + \frac{1}{x}\,\frac{d\,y^2}{d\,x^2} - \left(\frac{1}{x^2} + k^2\right)\frac{d\,y}{d\,x} = a\,x, \qquad (16)$$

where—

$$a = \frac{6\,p}{E\,t^3}(1-\mu^2) \text{ and } k^2 = \frac{12\,q}{E\,t^2}(1-\mu^2). \qquad (B)$$

This is easily reduced to Bessel's differential equation by putting—

$$\frac{d\,y}{d\,x} = \phi \text{ and } k\,x = v.$$

$$v^2\frac{d^2\phi}{d\,v^2} + v\,\frac{d\,\phi}{d\,v} - (1+v^2)\,\phi = \frac{a}{k^3}\,v^3. \qquad (16A)$$

Without writing out the complete integral of this equation, we will take a particular solution satisfying the condition necessary for us when—

$$v = k\,x = 0\;.\;.\;.\;\phi = 0.$$

This solution is—

$$\varphi = -\frac{k}{a^3} v + C\, J_1(v) \tag{17}$$

where $J_1(v)$ is Bessel's function, expressed by the series—

$$J_1(v) = \sum_{i=0}^{i=\infty} \frac{1}{1.2.3\ldots i.1.2.3\ldots i+1}\left(\frac{v}{2}\right)^{2i+1}$$

and C is an arbitrary constant, depending on the conditions of the securing of the perimeter of the plates.

As all the strains and stresses were expressed by us as derivatives of the function y of different orders; all the circumstances of the flexion may be considered found, except the strain q, which, as in the previous case of the rectangular plate, will be determined from the conditions of the displacement δx_0 of the contour of the plate in the direction of the radius—

$$\delta x_0 = \frac{1}{2}\int_0^a \varphi^2\, d x - \frac{q a}{E} \quad (1-\mu)\ ? \tag{18}$$

Having thus resolved the question in the general form, we may now pass to the particular cases.

(1) Real Plate.

$$\delta x_0 = 0. \quad (\sigma_x)_{x=a} = q.$$

This case corresponds to the absolutely stiff frame, where, however, the sections of the plate lying on the boundaries can freely incline as if on ideal hinges, so that for each point of the plate on the circular perimeter ($x = \pm a$)—

$$\sigma_x = q$$

or, by equation (14)—

$$\frac{1}{\mu}\frac{d^2 y}{d x^2} + \frac{d^2 y}{d z^2} = 0, \text{ with } x = \pm a.$$

From this condition the constant C is determined—

$$C = \frac{a}{k^2}\,\frac{1+\mu}{\left(\frac{d J_1}{d x}\right)_{x=a} + \mu\frac{J_1(k a)}{a}}$$

and thus—

$$\varphi = \frac{d y}{d x} = -\frac{a}{k^2} x + \frac{a}{k^3}\,\frac{1+\mu}{\left(\frac{d J_1}{d x}\right)_{x=a} + \mu\frac{J_1(k a)}{a}}\, J_1(k x) \tag{19}$$

In order to substitute this quantity in equation (18), let us expand it into a series of ascending powers of the "argument" $k\,x$; we shall obtain—

$$\varphi = \frac{a}{k^3}\left\{\frac{2 J_1(k x)}{8} - k x\right\}, \tag{19A}$$

where, for brevity's sake, we put—

$$k^2 a^2 = u,$$

and

$$s = 1 + \frac{u}{8} \cdot \frac{3+\mu}{1+\mu} + \frac{u^2}{8.24}\frac{5+\mu}{1+\mu} + \frac{u^3}{8.24.48}\frac{7+\mu}{1+\mu} + \ldots$$
$$+ \left(\frac{u}{4}\right)^{i-1} \frac{1}{(1.2.3\ldots\overline{i-1})^2\, i} \frac{2i-1+\mu}{1+\mu}$$

This series is convergent for all values of u. Substituting this quantity in the equation (18), we shall obtain, after integrations and reductions—

$$q = \frac{1}{2}E\left(\frac{a}{k^3 s}\right)^2 \left\{ \overline{s-1}^2 \frac{u}{3} - \frac{s-1}{4}\frac{u^2}{5} - \frac{2s-5}{4.48}\frac{u^3}{7} - \frac{s-7}{4.48.24}\frac{u^4}{9} - \frac{s-21}{4.48.24.80}\frac{u^5}{11} - \ldots \right\}.$$

Substituting instead of q, a, and k, their values from the relation (B), and instead of μ its numerical value for Martin steel (on the average about 0·3), we shall obtain an equation, one part of which contains only the "argument" u, and the other the known quantities, viz.:—

$$4{\cdot}3045 \ldots \left(\frac{p}{E}\right)^2 \left(\frac{a}{t}\right)^8 (1-\mu^2)^3$$

$$= 4{\cdot}3045 \frac{s^2 u^3}{216\left(\frac{1}{3}(s-1)^2 - \frac{1}{20}(s-1)u - \frac{2s-5}{192}\frac{u^2}{7} - \frac{s-7}{192.24}\frac{u^2}{9} - \ldots\right)}$$

$$= F_3(u). \qquad (20)$$

In the tables given below, a series of the value of $F_3(u)$ is calculated for different values of the "argument" u from 0 to 10. Having found u, and consequently, from relation (B), q, it is easy to find the extensions and strains for every point of the plate.

The maximum strain is obtained at the middle of the plate, and the corresponding stress is determined by the equation—

$$(E\,\epsilon)_{\text{max.}} = \sigma_{\text{max.}} = \frac{3}{8}p\left(\frac{a}{t}\right)^2(1-\mu)(3+\mu)\left[\frac{1}{7{\cdot}5}\frac{u}{\sqrt{F_3(u)}} + \frac{8(s-1)(1+\mu)}{s\,u\,(3+\mu)}\right]$$

$$= \frac{3}{8}p\left(\frac{a}{t}\right)^2(1-\mu)(3+\mu)f_3(u) \qquad (21)$$

The maximum deflection of the plate will be obtained by integrating the equation (19), and is equal to

$$y_{\text{max.}} = \frac{3}{16}\frac{p}{E}\left(\frac{a}{t}\right)^4 t(1-\mu)(s+\mu)\left[\frac{1+\mu}{5+\mu}\,\frac{16(s-1) - u - \frac{u^2}{36} - \frac{u^3}{36.64}\ldots}{s\,u}\right]$$

$$= \frac{3}{16}\frac{p}{E}\left(\frac{a}{t}\right)^4 t(1-u)(5+\mu)\varphi_3(u). \qquad (22)$$

The functions $f_3(u)$ and $\phi_3(u)$ are calculated from the same values of the "argument" u from 0 to 10; moreover,

$$f_3(0) = 1, \quad \text{and } \phi_3(0) = 1.$$

It remains to show that the series entering into the functions $F_3(u)$, $f_3(u)$, and $\phi_3(u)$ are convergent. The usual method would lead to very extensive numerical computations, as it is necessary to write down the general term of the series, expressed in a very complex manner. I prefer, therefore, to use the following practical reasoning:—

Let us calculate by the equation of the meridian the length l of its arc (from $x = 0$ to $x = a$) for the maximum value of the "argument" ($u = 10$), and, if it coincides with the length found from the relation

$$l = a\left(1 + \frac{q}{E}\right), \qquad \text{(C)}$$

then our series are practically convergent for the values of the "argument" u less than 10.

The equation of the meridian is obtained from equation (19A)—

$$y = \int_x^a \phi\, dx = y_{\max.} - \frac{3}{16}(1 - \mu^2)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\left(\frac{x}{a}\right)^2 \left[\frac{16(s-1)}{s u} - \frac{1}{s}\left(\frac{x}{a}\right)^2\left(1 + \frac{u}{36}\left(\frac{x}{a}\right)^2 + \left(\frac{u}{48}\right)^2\left(\frac{x}{a}\right)^4 + \right)\right]$$

Let us now calculate several ordinates y for different values of x, and from them find the perimeter inscribed as a broken line, which evidently must be less than the length of the arc. In the following table the perimeters l_k of these broken lines, for a different number of segments, are given. (See Plate II. Fig. 7.)

Number of Segments from 0 to a.	Ratio $l_k : a$, u being equal 10.
1	$1 + 0{\cdot}729\left(\frac{t}{a}\right)^2$
2	$1 + 0{\cdot}867\left(\frac{t}{a}\right)^2$
4	$1 + 0{\cdot}900\left(\frac{t}{a}\right)^2$

The length of the arc of meridian l, obtained from relation (C) is—.

$$l \neq a\left(1 + \frac{q}{E}\right) = a\left\{1 + \frac{u}{12(1-\mu^2)}\left(\frac{t}{a}\right)^2\right\} = a\left(1 + 0{\cdot}916\left(\frac{t}{a}\right)^2\right).$$

This last figure is sufficiently near to the previous one, and a little greater than it; this shows that the geometrical interpretation ascribed to our functions within the limits necessary for us is sufficiently exact.

(2) Real Plate.

$$\partial x_0 = 0. \qquad \phi_0 = 0.$$

This case corresponds to the absolutely stiff frame, and to a plate built into it so that its sections lying on the contour cannot become inclined.

The constant C of the equation (17) will be determined by the condition—

$$\phi_0 = 0 = -\frac{a}{k^3} u + C\,J_1(u),$$

and consequently the equation (17) will be written in the form—

$$\phi = \frac{a}{k^3}\left(\frac{\sqrt{u}\,J_1(kx)}{J_1(\sqrt{u})} - kx\right).$$

By substitution in equation (18) we find—

$$\frac{9}{35}\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8(1-\mu^2)^3 = \frac{u}{1 - \frac{5}{36}u + \frac{u^2}{69{\cdot}37} - \frac{u^3}{755{\cdot}5} + \ldots} = F_4(u), \qquad (23)$$

an equation serving to find u and also q, and solved by means of the table (from $u = 0$ to $u = 6$). As before we shall find the maximum deflection—

$$y_{max.} = \frac{3}{16}(1-\mu^2)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\left(1 - \frac{5}{72}u + \frac{11}{24.96}\mu^2 - \ldots\right) = \frac{3}{16}(1-\mu^2)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_4(u) \ldots \qquad (24)$$

and the stress corresponding to the greatest strain—

$$(E\,\epsilon)_{max.} = \sigma_{max.} = \frac{3}{4}p(1-\mu^2)\left(\frac{a}{t}\right)^2\left\{\frac{1}{24{\cdot}2}\frac{u}{\sqrt{F_4(u)}} + 1 - \frac{1}{24}u + \frac{1}{24.16}u^2 - \frac{1}{24.16.15}u^3 + \ldots\right\}$$

$$= \frac{3}{4}(1-\mu^2)\left(\frac{a}{t}\right)^2 f_4(u). \qquad (25)$$

This strain is obtained on the perimeter of the plate, and is almost twice that appearing in the middle of the plate. This relation is, however, true only so long as the maximum tension at the contour does not exceed the limit of elasticity, after which this tension passes to the middle of the plate, and the conditions of flexion nearly approach to those of the preceding case.

(3) Absolutely Stiff Plate.

$$u = 0.$$

In this case the stress q and the functions $F_3(u)$ and $F_4(u)$ become nothing, and the functions $f_3(u)$, $f_4(u)$, $\phi_3(u)$, and $\phi_4(u)$ become unity, for which reason the formulæ (21), (22), (24), and (25) may be applied to this case, if the factors f and ϕ in them be rejected. In this form the formulæ coincide with the ordinary formulæ for a circular plate, given in text-books on the theory of elasticity.

(4) Absolutely Flexible Plate.

$$u = \infty. \qquad \delta x_0 = 0.$$

The connection between p and q in this case will be determined from the condition—

$$\pi x^2 p + 2\pi x q t \frac{dy}{dx} = 0,$$

whence

$$\frac{d\,y}{d\,x} = \phi = -\frac{p\,x}{2\,q\,t};$$

the stress q will be determined from the condition—

$$\frac{1}{2}\int_0^a \phi^2\,d\,x = \frac{a\,q}{\text{E}} = \frac{1}{24}\left(\frac{p}{q}\right)^2\frac{a^3}{t^2},$$

or

$$q = \sqrt[3]{\frac{1}{24}p^2\,\text{E}\left(\frac{a}{t}\right)^2} \qquad (26)$$

and the maximum deflection of the plate—

$$y_{\max.} = \frac{1}{4}\frac{p}{q}\frac{a^2}{t} \qquad (27)$$

As these equations are distinguished from the analogous ones of the rectangular plate (3A) and (4) only by the constant multiplier, they can be generalised for the tensions q which pass the limit of elasticity of material. Denoting by ϵ the relative elongation of material corresponding to the tension q, we find—

$$p = 2\,q\,\frac{t}{a}\sqrt{6\,\epsilon} \qquad (28)$$

and—

$$y_{\max.} = \frac{p\,a^2}{4\,q\,t} = a\sqrt{\frac{3}{2}\epsilon} \qquad (29)$$

Thus we can obtain the formula for thin spherical shells subjected to internal pressure—

$$t = \text{R}\,\frac{p}{2\,q}.$$

If we wish to determine the immediate connection between the span $2\,a$, the thickness of the plate t, and the maximum tension $\sigma_{\max.}$, we can easily obtain—

$$\text{F}\,(u)\,[f\,(u)]^2 = \psi\,(u) = \kappa\left(\frac{\sigma_{\max.}}{\text{E}}\right)^2\left(\frac{a}{t}\right)^4, \qquad (30)$$

where κ is a numerical factor equal to 0·416 for the circle and rectangle built-in; this equation with the equations (9), (12), (21), and (25), can determine the limiting pressure p, corresponding to the given $\sigma_{\max.}$, and the ratio $\left(\frac{a}{t}\right)$.

If we denote by p_1, p_2, p_3, and p_4 the limiting pressures for—

(1) A rectangle whose one side is equal to $2\,a$, the other being very long;

(2) A rectangle with sides $2\,a$ and $2\,b$ ($b > a$);

(3) An ellipse with axis $2\,a$ and $2\,b$ ($b > a$);

(4) A circle with diameter $2\,a$ (see Fig. 3, Plate I.);

we have for very stiff plates built in on the boundaries the proportion—

$$p_1 : p_3 : p_4 = \frac{3}{8} : \frac{3}{8}\left(1 + \frac{2}{3}\left(\frac{a}{b}\right)^2 + \left(\frac{a}{b}\right)^4\right) : 1.^* \qquad \text{(K)}$$

Thus—

$$p_3 = 1{\cdot}23\, p_1, \text{ when } b = 2\,a, \text{ and}$$

$$p_3 = 1{\cdot}086\, p_1, \text{ when } b = 3\,a.$$

It is very probable that—

$$p_1 < p_2 < p_3,$$

and consequently p_2 is very near to p_1, and differs from it by less than 20 per cent., when $b > 2\,a$.

For the real plates we have—

$$p_1 : p_4 = \frac{3}{8}\frac{1}{f_2(u)} : \frac{1}{f_4(u)};$$

If the ratio of the span $2\,a$ to the thickness of plate t is 180, or less, and the ratio—

$$\frac{\sigma_{\max.}}{\mathrm{E}} = \text{nearly } \frac{1}{2000}$$

(as for Martin's steel), the fractions—

$$\frac{1}{f_2(u)} \text{ and } \frac{1}{f_4(u)}$$

are very near to unity (maximum 1·16).

Therefore it is probable that p_3 is included between the two following limits—

$$2\frac{\sigma_{\max.}}{(1-\mu^2)f_2(u)}\left(\frac{t}{a}\right)^2\left[1 + \frac{2}{3}\left(\frac{a}{b}\right)^2 + \left(\frac{a}{b}\right)^4\right]$$

and—

$$2\frac{\sigma_{\max.}}{(1-\mu^2)f_4(u)}\left(\frac{t}{a}\right)^2\left[1 + \frac{2}{3}\left(\frac{a}{b}\right)^2 + \left(\frac{a}{b}\right)^4\right].$$

These limits are sufficiently near, and do not differ more than by 5 per cent., and approximately we have—

$$p_3 = \frac{\sigma_{\max.}}{4(1-\mu^2)}\left(\frac{t}{a}\right)^2\left[1 + \frac{2}{3}\left(\frac{a}{b}\right)^2 + \left(\frac{a}{b}\right)^4\right]\left[\frac{1}{f_2(u)} + \frac{1}{f_4(u)}\right]. \qquad (31)$$

If for the real plates the inequality—

$$p_1 < p_2 < p_3$$

is true, p_2 will, as before, be very near to p_1, when $b > 2\,a$; thus—

$$p_2 < \text{ but near } \frac{\sigma_{\max.}}{2(1-\mu^2)}\left(\frac{t}{a}\right)^2\frac{1}{f_2(u)},$$

and roughly for the riveted plate, neglecting the weakening of the section by the rivet holes—

$$p_2 = 0{\cdot}55\,\sigma_{\max.}\left(\frac{t}{a}\right)^2 \qquad (32)$$

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* This remarkable result is due to Prof. Bryan. (*See* A. E. H. Love: "A Treatise on the Mathematical Theory of Elasticity," vol. ii. p. 199.)

TABLES OF FUNCTIONS

$F(u)$, $f(u)$ AND $\phi(u)$ FOR THE RECTANGULAR AND CIRCULAR PLATES.

A LONG RECTANGLE SUPPORTED ON THE BOUNDARIES $\delta x_0 = 0$. $M_0 = 0$.

u	$F_1(u)$	$f_1(u)$	$\phi_1(u)$
0·00	0·000	1·000	1·000
0·25	0·304	0·991	0·908
0·50	0·724	0·939	0·831
0·75	1·275	0·887	0·766
1·00	1·974	0·839	0·711
1·5	3·89	0·756	0·621
2·0	6·55	0·688	0·551
2·5	10·12	0·633	0·496
3·0	14·70	0·586	0·450
3·5	20·44	0·547	0·412
4·0	27·4	0·512	0·380
4·5	35·8	0·482	0·352
5·0	45·8	0·455	0·329
5·5	57·4	0·432	0·308
6·0	70·7	0·411	0·290
6·5	86·0	0·392	0·274
7·0	103·2	0·376	0 259
7·5	122·4	0·361	0·246
8·0	143·9	0·347	0·234
8·5	167·8	0·334	0.223
9·0	194·1	0·323	0·213
9·5	223·1	0·312	0·204
10·0	255	0·302	0·196
11·0	327	0·284	0·181
12	412	0·268	0·169
13	510	0·255	0·158
14	622	0·243	0·148
15	750	0·232	0·140
16	894	0·222	0·132
17	1,055	0·213	0·125
18	1,235	0·205	0·119
19	1,434	0·198	0·113
20	1,653	0·191	0·108

A CIRCLE SUPPORTED ON THE BOUNDARIES $\delta x_0 = 0$. $(\sigma_x)_{x=a} = q$.

u	$F_3(u)$	$f_3(u)$	$\phi_3(u)$
0·00	0·000	1·000	1·000
0·25	0·282	1·005	0·945
0·50	0·627	0·972	0·892
0·75	1·047	0·937	0·845
1·00	1·542	0·904	0·806
1·5	2·77	0·844	0·734
2·0	4·37	0·789	0·674
2·5	6·40	0·741	0·623
3·0	8·88	0·698	0·579
3·5	11·90	0·660	0·541
4·0	15·43	0·627	0·507
4·5	19·5	0·597	0·476
5·0	24·3	0·569	0·451
5·5	29·7	0·545	0·427
6·0	35·8	0·522	0·406
6·5	42·8	0·500	0·387
7·0	50·6	0·481	0·370
7·5	59·5	0·464	0·354
8·0	68·9	0·447	0·339
8·5	79·2	0·432	0·326
9·0	90·5	0·419	0·313
9·5	102·5	0·406	0·301
10·0	116	0·394	0·290

The corresponding formulæ are—

$$F_1(u) = \frac{1632}{35}(1-\mu^2)^3\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 = 35{\cdot}137\ldots\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 \qquad (7\text{B})$$

$$F_3(u) = 4{\cdot}3045\ldots(1-\mu^2)^3\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 = 3{\cdot}2437\ldots\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 \qquad (20)$$

$$q = \frac{Eu}{12(1-\mu^2)}\left(\frac{t}{a}\right)^2 = 0{\cdot}09157\,Eu\left(\frac{t}{a}\right)^2 \qquad (\text{A}_2)\text{ and }(\text{B})$$

$$\sigma_{\max.} = 3(1-\mu^2)p\left(\frac{a}{t}\right)^2 f_1(u) = 2{\cdot}73\,p\left(\frac{a}{t}\right)^2 f_1(u) \qquad (9\text{A})$$

$$\sigma_{\max.} = \frac{3}{8}(1-\mu)(3+\mu)p\left(\frac{a}{t}\right)^2 f_3(u) = 0{\cdot}866\,p\left(\frac{a}{t}\right)^2 f_3(u) \qquad (21)$$

$$y_{\max.} = \frac{5}{2}(1-\mu^2)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_1(u) = 2{\cdot}275\,\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_1(u) \qquad (8\text{B})$$

$$y_{\max.} = \frac{3}{16}(1-\mu)(5+\mu)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_3(u) = 0{\cdot}696\,\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_3(u) \qquad (22)$$

TABLES OF FUNCTIONS.

A Long Rectangle Built-in on the Boundaries $\delta x_0 = 0.\quad \phi_0 = 0.$

u	$F_2(u)$.	$f_2(u)$.	$\phi_2(u)$.	$\psi_2(u)$.
0·0	0·000	1·000	1·000	0·000
0·2	0·208	1·012	0·980	0·213
0·4	0·433	1·008	0·962	0·440
0·6	0·674	1·003	0·943	0·680
0·8	0·933	0·997	0·926	0·927
1·0	1·210	0·990	0·909	1·184
1·2	1·505	0·983	0·893	1·455
1·4	1·819	0·976	0·877	1·731
1·6	2·152	0·969	0·862	2·02
1·8	2·504	0·962	0·848	2·32
2·0	2·877	0·954	0·834	2·62
2·2	3·27	0·947	0·820	2·93
2·4	3·69	0·940	0·807	3·26
2·6	4·12	0·933	0·794	3·59
2·8	4·58	0·926	0·782	3·92
3·0	5·06	0·919	0·770	4·27
3·2	5·56	0·912	0·758	4·63
3·4	6·09	0·906	0·747	5·00
3·6	6·64	0·899	0·736	5·36
3·8	7·21	0·893	0·725	5·75
4·0	7·81	0·887	0·715	6·14
4·2	8·44	0·880	0·705	6·53
4·4	9·09	0·874	0·696	6·94
4·6	9·76	0·868	0·686	7·35
4·8	10·46	0·862	0·677	7·76
5·0	11·19	0·857	0·668	8·20
5·2	11·95	0·851	0·659	8·65
5·4	12·73	0·846	0·651	9·11

A Circle Built-in on the Boundaries $\delta x_0 = 0.\quad \phi_0 = 0.$

u	$F_4(u)$.	$f_4(u)$.	$\phi_4(u)$.	$\psi_4(u)$.
0·0	0·000	1·000	1·000	0·000
0·2	0·206	1·010	0·986	0·210
0·4	0·423	1·009	0·973	0·430
0·6	0·652	1·007	0·960	0·661
0·8	0·892	1·003	0·948	0·897
1·0	1·143	1·000	0·935	1·143
1·2	1·408	0·995	0·923	1·391
1·4	1·685	0·991	0·911	1·655
1·6	1·975	0·986	0·900	1·92
1·8	2·278	0·981	0·889	2·19
2·0	2·594	0·977	0·878	2·47
2·2	2·923	0·972	0·867	2·76
2·4	3·27	0·968	0·857	3·06
2·6	3·63	0·963	0·847	3·36
2·8	4·00	0·959	0·837	3·58
3·0	4·38	0·954	0·828	3·98
3·2	4·78	0·950	0·818	4·31
3·4	5·19	0·945	0·808	4·64
3·6	5·62	0·940	0·799	4·97
3·8	6·07	0·936	0·790	5·32
4·0	6·53	0·932	0·781	5·68
4·2	7·00	0·927	0·773	6·02
4·4	7·49	0·922	0·765	6·36
4·6	8·00	0·917	0·757	6·73
4·8	8·52	0·913	0·750	7·10
5·0	9·05	0·909	0·742	7·48
5·2	9·61	0·905	0·735	7·87
5·4	10·19	0·902	0·728	8·29

The corresponding formulæ are—

$$F_2(u) = \frac{64}{35}(1-\mu^2)^3\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 = 1{\cdot}378\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 \qquad (10\text{B})$$

$$F_4(u) = \frac{9}{35}(1-\mu^2)^3\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 = 0{\cdot}1937\left(\frac{p}{E}\right)^2\left(\frac{a}{t}\right)^8 \qquad (23)$$

$$q = \frac{E\,u}{12(1-\mu^2)}\left(\frac{t}{a}\right)^2 = 0{\cdot}09157\ \ldots\ E\,u\left(\frac{t}{a}\right)^2 \qquad (\text{A}_2)\text{ and }(\text{B})$$

$$\sigma_{\max.} = 2(1-\mu^2)\,p\left(\frac{a}{t}\right)^2 f_2(u) = 1{\cdot}82\,p\left(\frac{a}{t}\right)^2 f_2(u) \qquad (12\text{A})$$

$$\sigma_{\max.} = \frac{3}{4}(1-\mu^2)\,p\left(\frac{a}{t}\right)^2 f_4(u) = 0{\cdot}682\,p\left(\frac{a}{t}\right)^2 f_4(u) \qquad (25)$$

$$y_{\max.} = \frac{1}{2}(1-\mu^2)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_2(u) = 0{\cdot}455\,\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_2(u) \qquad (11\text{A})$$

$$y_{\max} = \frac{3}{16}(1-\mu^2)\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_4(u) = 0{\cdot}171\,\frac{p}{E}\left(\frac{a}{t}\right)^4 t\,\phi_4(u) \qquad (24)$$

$$\psi_2(u) = F_2(u)\,[f_2(u)]^2 = \frac{16}{35}(1-\mu^2)\left(\frac{\sigma_{\max.}}{E}\right)^2\left(\frac{a}{t}\right)^4 = 0{\cdot}416\left(\frac{\sigma_{\max.}}{E}\right)^2\left(\frac{a}{t}\right)^4 \qquad (30)$$

$$\psi_4(u) = F_4(u)\,[f_4(u)]^2 = 0{\cdot}416\left(\frac{\sigma_{\max.}}{E}\right)^2\left(\frac{a}{t}\right)^4 \qquad (30)$$

EXAMPLE 1.—A steel circular plate, whose thickness $t = \frac{1}{4}$ in., and radius $a = 16$ in., is subjected to a uniform pressure of $p = 10$ lbs. per square inch. The edges of the plate are so fastened to the supporting frame that they cannot shift horizontally ($\delta x_0 = 0$), but the edge elements are free to incline (σ_x on the boundaries is equal q); the Young's modulus $E = 33,300,000$ lbs. per square inch. We have—

$$F_3(u) = 3{\cdot}244 \ldots \left(\frac{p}{E}\right)^2 \left(\frac{a}{t}\right)^8 = 82{\cdot}2. \qquad (20)$$

Thus, in the first and second columns of the corresponding table we find—

$$u = 8{\cdot}63,$$

and consequently—

$$f_3(u) = 0{\cdot}429\,; \quad \phi_3(u) = 0{\cdot}323.$$

The maximum stress—

$$\sigma_{max.} = 0{\cdot}866\, p \left(\frac{a}{t}\right)^2 f_3(u) = 35{,}470 \times 0{\cdot}429$$

$$= 15{,}220 \text{ lbs. per square inch} = 6{\cdot}8 \text{ tons per square inch.} \qquad (21)$$

The maximum deflection—

$$y_{max.} = 0{\cdot}696\, \frac{p}{E}\left(\frac{a}{t}\right)^4 t\, \phi_3(u) = 0{\cdot}876 \times 0{\cdot}323 = 0{\cdot}283 \text{ in.} \qquad (22)$$

The same values obtained by the ordinary formulæ for stiff plates will be correspondingly 15·8 tons per square inch and 0·876 in.

EXAMPLE 2.—A steel rectangular plate, whose thickness $t = \frac{9}{20}$ in., breadth $2\,a = 50$ in., and length is very great (practically is more than 10 to 12 ft.), is subjected to the pressure of a water column the height of which is 18 ft. The edges of the plate are built in ($\phi_0 = 0$), and cannot shift horizontally ($\delta x_0 = 0$).

Thus we have—

$$p = 18 \times \frac{4}{9} = 8 \text{ lbs. per square inch.}$$

$$F_2(u) = 1{\cdot}378 \left(\frac{p}{E}\right)^2 \left(\frac{a}{t}\right)^8 = 7{\cdot}21, \qquad (10\text{B})$$

and consequently—

$$u = 3{\cdot}80, \quad f_2(u) = 0{\cdot}893, \quad \phi_2(u) = 0{\cdot}725.$$

$$\sigma_{max.} = 1{\cdot}82\, p \left(\frac{a}{t}\right)^2 f_2(u) = 40{,}300 \text{ lbs. per square inch} = 18 \text{ tons per square inch.} \qquad (12\text{A})$$

$$y_{max.} = 0{\cdot}455\, \frac{p}{E}\left(\frac{a}{t}\right)^4 t\, \phi_2(u) = 0{\cdot}34 \text{ in.} \qquad (11\text{A})$$

We cannot admit such a stress exceeding the elastic limit of the materials. If we wish to obtain the limiting pressure on this plate, we can write, making, for example, $\sigma_{max.} = 8$ tons per square inch $= 17{,}920$ lbs. per square inch—

$$\psi_2(u) = 0{\cdot}416 \left(\frac{\sigma_{max.}}{E}\right)^2 \left(\frac{a}{t}\right)^4 = 1{\cdot}15, \qquad (30)$$

and consequently—

$$u = 0{\cdot}976 \text{ and } p = 0{\cdot}55\, \sigma_{max.} \left(\frac{t}{a}\right)^2 \frac{1}{f_2(u)} = 3{\cdot}2 \text{ lbs. per square inch,}$$

or the limiting height of water column is 7·2 ft.

DISCUSSION.

Mr. Francis Elgar, LL.D., F.R.S. (Vice-President): My Lord, might I say a word upon this paper before Professor Bryan speaks? The paper I read several years ago upon the strength of bulkheads, which is referred to by Monsieur Boobnoff, merely gave reasons to show why I did not see my way to believe in any quantitative solutions which had, up to that time, been put before us of the problems that the author here attempts to solve. It had, at any rate, one good effect, and that was to induce Professor Bryan, who is here to-day, and who is perhaps better able to express an opinion upon these very difficult problems than anyone else in the room, to apply his mind to the subject, and to favour us with the results of his investigations. I hope he will speak upon the present paper; but, before he does so, I would call attention to one or two points that strike one at a hasty glance over this paper. It would be impossible for me to attempt to express an opinion right off upon such difficult mathematics as are dealt with here; but one or two results are arrived at and described that appear to require attention. An example of these will be found in the first and second paragraphs of page 23. Monsieur Boobnoff there states, as the result of his investigations, that "with a thickness of outer shell plating of a large battleship of $\frac{16}{20}$ of an inch" (in our large battleships it is not so much as $\frac{16}{20}$ of an inch, it is only $\frac{12\frac{1}{2}}{20}$), "and with a space between the transverse frames of 48 in., the admissible pressure is approximately equal to 24 ft. of salt water." Now the pressures which our battleships ordinarily have to bear in practice reach as high as that due to a head of 28 ft. of salt water at the lowest part of the plating, with a thickness of shell plating which is less by 22 per cent. than that given in the paper.

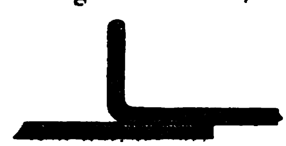
Mr. Archibald Denny: That is in smooth water?

Dr. Elgar: That is in smooth water with the vessel at rest, when there is simply the steady pressure due to the head of water that is denoted by the draught. It appears to me, therefore, that investigations leading to this result (which is not on the face of it), consistent with experience, require very careful examination from all points of view, before they can be accepted as conclusive. I am sure we should all be grateful to Professor Bryan for his opinion. Whether Monsieur Boobnoff has succeeded in absolutely solving a problem that has so long presented insuperable difficulty, or whether he has come somewhat short of that, there can, however, be no question that he has brought before us a most instructive investigation of the resistance of thin plates to flexure, and one that marks a great advance in treatment and in power of approximation to correct results. Its full value can only be judged after close study of all the points that are discussed, and full consideration of these difficult and interesting investigations and the practical results to which they lead.

Professor G. H. Bryan, M.A., F.R.S. (Visitor): My Lord and Gentlemen, Monsieur Boobnoff's paper makes me recall the difficulties which I myself saw looming ahead in an endeavour to apply the mathematical theory of the thin plate to calculating the stresses in the kind of plating which has to be dealt with in ships. If it should appear that the results found in the paper do not agree with those which Dr. Elgar has found to exist in practice, that is no reason for discontinuing the study of the question from the mathematical standpoint. What is necessary is to find out why they do not agree, and where the investigation can be improved. Monsieur Boobnoff has gone farther than previous workers, starting from the mathematical point of view, in taking both tensile and bending strains into account simultaneously, in the case of the strip which he calls the long rectangular plate and in that of the circular plate. Now, in all probability, both those kinds of strains exist simultaneously in actual plating. The rigidity for bending strains only is proportional to the cube of the thickness, whereas the rigidity for tensile strains is proportional to the first power of the thickness. On the other hand, the tensile strains are proportional only to the square of the displacement of the middle surface of the plate, while the bending strains are proportional to the first power of the displacement. Now, Monsieur Boobnoff makes use of certain postulates. He admits that the rectangle, both of whose sides are finite, is a case which does not admit of *simple* mathematical solution, and I can confirm the statement that the solution, when found, is one "whose convergency leaves much to be desired for practical calculation." But I have been in very great doubt as to whether the postulate which he assumes could be admitted. I saw that the case of an ellipse with built-in sides could be investigated both for a uniform pressure and for a hydrostatic pressure varying as the depth below any line. The objection to applying the postulate appears to me to arise from the corners of the rectangle. The distribution of stress in a plate having sharp angles may be very different to that in one with a rounded contour, or in a strip where the effects of the ends are neglected, and where the plating can bend without any tensile stress being produced in it if the supports can give way. A rectangular plate cannot be bent to more than a certain extent without tensile stresses being produced in it. A strip, on the other hand, can bend into a cylinder. To a mathematician, the fact that the solution of the problem of the bent rectangular plate involves the use of Fourier's Series, "whose convergency leaves much to be desired for practical calculation," suggests the idea that there may be a very great tendency to break at the corners of the rectangle, if the series fails to be convergent at these points. Now, what appears to me to be required for the further study of the stresses in plating is this:—The mathematician wants to know not so much the actual stresses in the plate, as the shape of the plate when it is distorted by pressure, and, if he cannot find that mathematically, I believe it would be possible to do so experimentally by some such method as the following:—Let a rectangular plate be taken, let one of its faces be dotted over with little round glasses silvered on the back, and let these mirrors reflect light from a suitable source on to a screen. Then let the plate be submitted to pressure on the opposite face, the consequence will be that it becomes distorted, and the patches of light reflected by the mirrors will be displaced, their displacements determining the angles through which the surface of the plate is bent at each point. Once these angles have been found, it is an easy matter of integration or quadrature to find the actual displacement of the plate at every point, and, on the other hand, it will be very easy to calculate the stresses at different points. What I should suggest is that, when the curves of displacement have been found, a contour map should be drawn of the displaced plate. This should be done at several different pressures. If the bending stresses alone predominate, the displacement at every point will be proportional to the pressure, and the *form* of the contour lines on the map will be independent of the pressure. On the other hand, any change in the form of the contour lines when the pressure is increased will indicate that tensile as well as bending strains have been brought into play, and a measure will thus be obtained as to the extent to which the actual plate differs from the plate of Kirchhoff's and other mathematical investigations, where bending and not tensile strains are taken into account. Such experiments may at first sight appear to be rather of mathematical than of practical interest, but I consider it is of the greatest importance that we should endeavour to learn what stresses exist in a rectangular plate, and especially whether the corners are strong or weak parts of the plate. In conclusion, it is a matter of great importance that contour maps of the displaced forms of rectangular plates should be drawn under varying pressures, and I would invite the members of this Institution to make experiments for the purpose. As a further inducement, may I point out that when once the necessary apparatus has been provided, it will afford a ready means of detecting flaws in plating, since any such flaw must of necessity give rise to irregularities in the contour lines of the plate when bent under pressure.

Mr. J. Bruhn, D.Sc. (Member): My Lord and Gentlemen, I think this is a very interesting paper; it deals with the question in the right way, both from the point of calculation and from experiment, and I have no doubt that Monsieur Boobnoff's explanation of the stresses is right. It has always been rather a wonder how the frames could be spaced so wide apart as they are in warships, that is to say, 48 in., because, by the ordinary calculations, the stresses would be far too high. I suppose the proposed method of calculation, assuming that a catenary tension exists, really points to the right explanation of the possibility of such wide frame spacing. It is highly desirable, as Monsieur Boobnoff has pointed out, that a correct method should be devised, whereby the spacing of the frames can be determined on some right principle, particularly where new departures are made, or where new features are introduced in designs, or else where old features or old designs are modified in some respects, as, for instance, when the size of the ship, or the proportion of length to depth are increased. Of course, the tables given in this paper apply entirely to warship practice—that is, to such wide frame spacing as 48 in. I think it is a pity that some calculations were not made for a closer frame spacing, something like that used in the mercantile service—say, half of that given, or 24 to 30 in. It would have been particularly interesting if Table C had been extended so as to take in proportions of spacing to thickness of say, 30, 40, and 50, and the corresponding results given. I do not know whether Monsieur Boobnoff has any figures for those proportions; if so, it would, I think, be very interesting to have them. In the merchant service, the stresses are probably somewhat higher on the bottom than in the case of warships. Merchant vessels are longer and the compressive stresses would, therefore, probably be higher, and for that reason it is necessary to have closer frame spacing. The stresses calculated by the method proposed in this paper are simply those due to the buckling or bending of the plates, and to those must be added the compressive stresses due to hogging. I should rather think that when the vessel is hogging, the worst case would happen with regard to the frame spacing. The pressure amidships would be somewhat greater than when the vessel is sagging, and, in addition, the bottom plating would be in compression. I am not quite sure whether these proposed calculations would apply then, because it seems to me that, when the bottom plating is in compression, there can be no catenary tension, and it cannot consequently take part of the stresses. The absence of any tension would seem to make the proposed method of calculation inapplicable. I have not been able to read through the whole of the Appendix, and perhaps there is an explanation of this point there; but, if not, I should like to know if Monsieur Boobnoff has made any calculations on the assumption that the plating is under strong compression, while at the same time subjected to water pressure. There is another point: If the plating is in tension, it would probably be right simply to add this additional stress to that caused by the pressure of the water, but if the plating is in compression, there would be an additional tendency to buckle, that is to say, the plating would buckle even apart from any water pressure, or, at any rate, it would not give way simply from compressive stresses. Apparently, this tendency has been omitted in arriving at the total stress on the plating. In proposing the Z bars between the floors, I see that the strength of those have been calculated on the ordinary assumption, that is to say, by the bending moment formulæ. In that case, of course, the bars would yield to a considerable extent, and that would have some effect on the plating; I do not know whether that has been included in the calculations. The proposed method of practical construction is no doubt the correct one from a theoretical point of view. Weight will always be saved by having deep floors and web frames with continuous fore and aft angles or Z bars between, but it is a question whether this is economical from the financial point of view, as the method would entail a large amount of riveting. I should like to know from Monsieur Boobnoff how far these calculations could be made applicable to surfaces supported by girders; not simply plain plates, but supported by stiffeners, say, at right angles to each other, such as would be the case at the side of the ship when the frames, web frames, and stringers are included. I should think that, in that case, the ordinary bending moment formulæ would be practically correct in all cases, where the supported area is of an oblong form, as is usually the case.

Professor J. H. BILES, LL.D. (Member of Council): I am afraid, Gentlemen, that, at this hour, it is hardly desirable to take up your time with many remarks on this subject, but I had in my mind exactly the same idea, after reading this paper, as that which Professor Bryan has expressed, and that is, in the first place, that the results are very startling, and, in the second place, that it is exceedingly desirable that the results should as soon as possible be tested by experiment. That is the practical thing to do in such a case. We had, I think, before, when Professor Bryan read his paper, a reference of this subject to a committee to test experimentally the result of putting thin plates under pressure, and, if my memory is correct on this matter, I think that Lloyd's undertook to carry out these experiments in conjunction with Professor Bryan and Dr. Elgar. I do not know whether these experiments were ever begun, but I do not think they were ever completed; at any rate, we have never had any results from them. If we could induce those gentlemen to take up that subject again, with the further light that Monsieur Boobnoff has thrown on the question, we may, perhaps, this time get some practical results from it. With respect to the structure that is proposed, of course, if these figures are correct—and there is a great deal of doubt about their accuracy—the character of the results may be correct, but they may be generally too high. If there is any great stress to be laid upon this, there are many ways in which a bottom can be stiffened to resist pressure. Whether the proposed method is the best way or not is, of course, a matter for individual consideration in each case. Some fifteen or sixteen years ago, in connection with some light torpedo boats that I designed, the question of the thickness of the outside bottom was considered in relation to this pressure, and, in order to make things a little safer, we turned up the flanges of the shell plates, so that they were put on thus—



or the inside strake was, and the outside strake was fastened on so that the flanges of the plates were turned up and formed a stiffening, somewhat of the character that is proposed by the Z framing described in Monsieur Boobnoff's paper. This flange, of course, had to be cut in the case where we applied it, because the framing was connected direct to the shell, but it was carried close up from frame to frame. The flange was turned up for a certain distance, and naturally afforded a considerable support for that length of the plate. There is one thing with reference to what Mr. Bruhn has said, that in merchant ships, which are much longer than warships, the stresses due to tension and compression on a wave are probably much higher, and the necessity for having closer frame spacing then comes in, because, if you had the frame spacing as wide as in a warship, you might have these stresses, due to outside pressure, in addition to the main structural stresses. No doubt that is of some importance, and probably the reason that some of the long warships, such as the *Powerful* and the *Terrible* (which are as long as some of our merchant ships), do not show signs of weakness in the outer bottom is that they have sheathing outside, and that perhaps may be some argument in favour of retaining sheathing in some of the longer and more powerful high-speed cruisers.

The Right Hon. the Earl of GLASGOW, G.C.M.G., LL.D. (President): Gentlemen, I presume you will agree that the author be thanked by letter for his very interesting paper, as he is, unfortunately, not present.

The following additional contribution to the discussion has been received from Dr. F. ELGAR, LL.D., F.R.S. (Vice-President):—"I am sorry I was obliged to leave before the discussion was closed, and did not hear the remarks of Professor Biles upon what took place several years ago, with regard to experiments upon the flexure of thin plates under pressure. In the discussion on my paper upon the strength of bulkheads, at the Spring Meeting in 1893, it was suggested that the Council should be asked on behalf of the Institution to consider a scheme of experiments. This was not, however, acted upon, and no committee was appointed to deal with the question. I commenced to work at it, in conjunction with Mr. Martell, and Lloyd's Register Society voted a sum of money towards the cost. We got the length of making some preliminary experiments for the purpose of determining the number and nature of the whole series of experiments that would be required, and the best method of analysing and recording the results, but we found that it would require more time than we had at command to carry the scheme through, and we were reluctantly compelled, on that account, to abandon it. No action was taken by the Institution in the matter."

The following reply to the discussion on his paper has been received from Monsieur Ivan G. Boobnoff:—

The final practical deduction of my paper, that the stresses in the ship's bottom plating due to water pressure are not insignificantly small as is usually assumed, but, on the contrary, are extremely great, and frequently exceed all limits admissible in engineering, is in such contrast with the general opinion that I could only expect the most sceptical reception for my investigation. The science of naval construction, as a result of its historical development, has now arrived at such a point that "how it is done" carries infinitely more conviction than "how it should be done," and I have, therefore, no reply to Dr. Elgar's remarks, in which he points out that much thinner plating is made in practice than my calculations require. When it is made thinner than necessary, the stresses will merely be greater than necessary, and that is all. Professor Bryan expresses doubt as to the truth of my postulate (see § 7, p. 22) for a rectangular plate in which the length of one side exceeds the other by twice or more, as we know nothing of the magnitudes of the stresses on the corners of the rectangle, where they may be, as he thinks, very great. I arrive at this postulate by the following considerations: Let us compare the bending of such a plate with that of an ordinary beam; the difference will be, firstly, in the plate resisting, partly by bending and partly by stretching, and, secondly, in the plate being supported by the short sides. The first circumstance, for plates built-in on the boundaries, affords a very small advantage, not exceeding, as appears from the paper, fifteen per cent. for plates of Siemens-Martin steel, when the ratio of span to thickness is 180 or less. It may be further observed that this difference will be reduced to zero if the supports give way in the smallest degree; for example, it is sufficient that the frames of a large ship (frame spacing being 48 in.) should give way $\frac{1}{400}$ of an inch (less than the thickness of the paper used for printing the Transactions of this Institution), for all the catenary tensions to vanish, and the above-mentioned advantage with them. The support given by the short sides, evidently, also diminishes the stresses at a point situated at the middle of the long side (where, I consider, the stresses have their maximum value), but, of course, in inverse proportion to the length of the side of the rectangle. For a built-in elliptical contour, as Professor Bryan himself has shown, the stress at the ends of the short axis is almost independent of the length of the longer axis, and it follows, therefore, that the same can be assumed for the case of a rectangle. Thus, the difference between a beam built-in at both ends, and the above-mentioned rectangular plate may be disregarded in practical calculation. I would further point out that, for a practical man, it is of no importance to know exactly the maximum stress in his construction. In our case, we need only the approximate conviction that this stress is about eight tons per square inch, or more (with an error of about $\pm$ 10 per cent.), because, in a rationally constructed hull we cannot admit stresses due to water-pressure of more than 2 to 3 tons per square inch, as there also exist additional stresses, due to pitching and many other causes. I shall, therefore, not contest Professor Bryan's opinion that the maximum stress for a rectangular plate of finite size may be in the corners, although I do not fully agree with it. In any case, this circumstance cannot diminish the stresses calculated by me at the middle of its greater side, and they come out very high. As to Professor Bryan's proposal to make experiments on the bending of plates, it has my complete sympathy; but it must be borne in mind that, in observing by means of mirrors the angles of inclination of the various line-elements of the plate, in order to obtain the stresses sought for, we must differentiate the resulting curves of the observed angles, because the stresses in the plate depend on its curvature, and this can only be done satisfactorily when these angles are observed with very great exactitude (for example, down to 5 to 10 seconds), which would be very difficult to realise when the experiments are made upon a large scale. It is, therefore, simpler to carry out experiments of a comparatively rougher character, for rectangular and circular plates, with the object of verifying the above-mentioned formulæ and the fundamental postulate. It would be especially simple to observe, for various contours, the limiting pressure causing a permanent set on the boundaries, when the angle of inclination of the elements lying at the middle of the greater side abruptly increases. Of course, it is necessary to take care to avoid the giving way of the boundaries by the means indicated in Fig. 4, Plate I., or otherwise. I venture to hope that the results of these experiments (if made) will be published, as they offer great scientific interest. Mr. J. Bruhn regrets that I have devoted most of my attention to warships and have said little about those of the Mercantile Marine. The reason for this is that, in merchant ships, the frames are placed closer than in ships of war, and, therefore, the calculation of the stresses in the plating can be made very simple, using the formulæ for absolutely stiff plates, if only the intermediate frames can be considered to be equally strong supports as deep frames. But, as in practice their stiffness is considerably less than is necessary on this supposition, I was afraid that the discussion of this question would only complicate the problem before us, already sufficiently difficult in itself. I have, therefore, confined myself to the brief remarks, without full proofs, given at the end of § 8 (page 23). The figures of Table C, given in the paper, relate to those proportions of span to thickness for which the calculations of the limiting pressure require a great amount of work; but when this proportion diminishes, the calculations by the formulæ of the paper itself become more and more easy, and it is therefore very simple to continue the Table in this direction if necessary. The stresses due to hogging and sagging, referred to by Mr. Bruhn, are added algebraically to the stresses due to water pressure, and do not alter the character of the phenomenon. As to his question regarding the method of calculating the stresses in a plate supported by any desired stiffening ribs, such a calculation may be made by the above-mentioned formulæ, provided that the stiffness of these supports is sufficiently great, as compared with that of the plate itself. Professor Biles considers my figures very startling, and suggests that there is a great deal of doubt about their accuracy. I do not know what my critic means by "accuracy," and whether this word relates to the postulate and principle which were used to obtain the formulæ, or to the formulæ themselves, as the real expression of the principles involved, or to the reduction of the formulæ to numbers. As to the principles and postulates, I have given my explanation in my reply to Professor Bryan's remarks. The correctness of the formulæ and their results can easily be verified. As to the method employed by Professor Biles fifteen years ago for stiffening the plates of a torpedo boat by means of turned-up flanges, it is necessary to observe that this method certainly could not attain its object, as these flanges are cut by the frames, precisely at the places where they are most needed. The number of methods for stiffening the bottoms of ships are very numerous. The whole difficulty consists in avoiding useless increase of weight of hull and complication of work. A suitable method will, no doubt, be found as soon as practical ship-builders recognise the necessity of avoiding systems of construction of hull in which stresses in the bottom plating can exceed the limits admissible in all other branches of engineering.

ON LIQUID FUEL FOR SHIPS.

By Sir FORTESCUE FLANNERY, M.P. (Member).

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 20, 1902; the Right Hon. the Earl of GLASGOW, G.C.M.G., LL.D., President, in the Chair.]

THE subject of this paper is not in any sense new. The use of liquid fuel has been known for many years to be mechanically possible, and the Transactions of this Institution contain many important references to the question. The late Admiral Selwyn was an enthusiastic advocate and an eloquent exponent of the special advantages of liquid fuel for vessels of war.

The use of this fuel on the Caspian Sea and the Volga began in 1870, and is quite the recognised custom in about 200 vessels on the Caspian, and 200 on the Volga. The fact that the vessels navigate in fresh water on the Volga, and can, therefore, afford an unlimited supply of steam for pulverising the fuel without any risk of undue incrustration to their boilers, is an important factor. The Russian law, however, prohibits the exportation of liquid fuel of low flash point, and the difficulty of internal transport has been the dominant factor in the confinement of the use of liquid fuel at sea to the Caspian for at least a generation after its successful use there has been demonstrated.

Whilst the advantages of liquid fuel, and the possibility of its successful mechanical use, have been generally admitted, little or no progress in its application had been made outside the Russian inland sea just referred to, and the reason of this stagnation has been mainly of a commercial character. The supply of fuel outside Russia has been but nominal, and no general application was possible, unless both war and mercantile vessels could be assured of continuous supply from year to year, and unless that supply were as regularly accessible at as frequent and convenient oiling stations throughout the world as already exist in the case of coal fuel, and at a cost proportionately as low. Such a condition of things never became possible until the recent discovery of large supplies of oil suitable for fuel, first in Borneo and Burmah, and quite recently in Texas and California. It is to be regretted that the only one of these sources of supply that lies in British territory is that of Burmah.

The whole aspect of the question, whether regarded by the Admiralty, the shipowner or the naval architect, has been changed by the assurance of continuous supplies of liquid fuel, and it becomes necessary to treat the question, not only as of practical importance, but of urgency to those responsible for the highest efficiency of fighting and carrying ships. The British Admiralty has determined to exhaustively test the use of this newly resuscitated means of evaporation, and the reference to the question by the First Lord in his recent Memorandum is a clear indication of progressive policy—a policy which is understood to extend to trials not only in destroyers, but also in three cruisers and one battleship. The Italian Admiralty have been pursuing the question for some years, even before large supplies were assured. The German Admiralty have used liquid fuel on the China station for many months in lieu of coal for auxiliary purposes on board ship. The Hamburg-American S.S. Company have fitted four steamers for liquid fuel, and the North German Lloyd two vessels. The Dutch Navy have fitted liquid fuel apparatus in conjunction with coal to two destroyers and Dutch mail and cargo steamers in the Far East have the new liquid fuel in regular use. Danish shipowners have ordered the building in Germany of two steamers to burn liquid fuel; and some twenty vessels under the British flag are now running regularly with liquid fuel; whilst at least a dozen are building with suitable fuel apparatus included in their design.

Supply.

Storage and supply of this fuel are in regular currency, or in course of arrangement, at the following ports:—London, Barrow, Southampton, Amsterdam, Copenhagen, New Orleans, Savannah, New York, Philadelphia, Singapore, Hong Kong, Madras, Colombo, Suez, Hamburg, Port Arthur, Texas, Rangoon, Calcutta, Bombay, Alexandria, Bankok, Saigon, Penang, Batavia, Surabaya, Amoy, Swatow, Fuchow, Shanghai, Hankow, Sydney, Melbourne, Adelaide, Zanzibar, Mombassa, Yokohama, Kobe, and Nagasaki; and storage arrangements are projected in South African and South American ports.

It may be expected that the supply to these stations will be drawn, as regards the ports east of the Suez Canal, from Borneo and Rangoon and, as regards those west of the Canal and in South America, from the Texas fields; South African stations being neutral as regards the heavy charges of the Suez Canal, are therefore likely to draw their supply from Borneo or Texas with equal economy. The South American stations will no doubt be supplied from the Texas and California fields.

Chemical Composition.

It is not within the scope of this paper to discuss exhaustively the refining and the chemical constituents of the fuel, but a general statement of what is meant by liquid fuel is necessary.

The product, as it issues from the wells, may be roughly said to be divisible into five principal sections, *i.e.*, spirit (benzine or naphtha); kerosene, or petroleum lamp oil; solar oil; astatki, or residuum, and water. The spirit, and the lamp and solar oils are too valuable commercially to be used as fuel in competition with coal; but the residuum, a thick treacle-like sluggish liquid is, if sufficiently separated from water and other impurities, very suitable for fuel, consisting, as it does, of the following chemical elements:—Carbon, 88 per cent.; hydrogen, $10\frac{3}{4}$ per cent.; oxygen, $1\frac{1}{4}$ per cent.

The two impurities whose presence is most inimical to steam generation are water and sulphur. Water puts out the flame and causes an inrush of cold air very injurious to the boiler, and sulphur, if free (that is not chemically combined with the oil), is injurious both to copper and steel. These impurities are now, however, of rare occurrence, and although, as will be seen later on, mechanical means of separation from water are necessary on board ship, this necessity arises rather from the admixture of ballast water than from water not taken out of the oil in the process of refining.

The question of safety and flash point is of extreme practical importance. The British Admiralty have hitherto required a flash point of 270° Fahr. Lloyd's Register has required a flash point of 200° Fahr., now reduced to 150°, whilst the German authorities have accepted as safe a flash point of 150° Fahr. Fuel of the lower flash point has been in constant use for four years in British and Dutch mercantile vessels with complete immunity from accident. It is not desirable to fix a flash point higher than is really necessary for safety, because high flash points are obtained by refining the volatile elements out of the liquid to such an extent as to leave a thick and sluggish residuum requiring large power to pulverise it into spray before ignition in the furnace, and necessarily more expensive by reason of the additional refining process.

Comparative Advantages and Disadvantages for War Vessels.

The problem that confronts every designer of a warship is the combination of the greatest speed, armament and ammunition supply, protection, and range of action, in the smallest and least expensive hull, and any reduction of weight and stowage room necessary to any of these qualities is a saving which acts and re-acts favourably upon the problem in a manner familiar to us all. The practical figures of comparison between coal and oil fuel realised in recent practice are that two tons weight of oil are equivalent to three tons weight of coal, and 36 cubic feet of oil are equivalent to 67 cubic feet of coal as usually stored in a ship's bunkers; that is to say, if the change of fuel be effected in an existing war vessel, or applied to any design without changing any of the data other than those affecting the range of action, the latter is increased by 50 per cent. upon the bunker weight allotted, and nearly 90 per cent. upon the bunker space allotted.

The coal protection for cruisers, whatever its real advantages—a matter upon which different opinions exist—would disappear with the use of liquid fuel, because it would be for the most part stowed below the water-line, if not wholly in the double bottom. The double bottom and other spaces, hitherto quite useless except for water stowage, would be capable of storing liquid fuel, and the space now occupied by coal bunkers would be available for other uses.

The ship's complement would be reduced by the almost complete abolition of the stoker element and the substitution of a limited number of men of the leading stoker class to attend to the fuel burners under the direction of the engineers, and the space of stokers' accommodation, the weight of their stores, together with the expense of their maintenance would be saved. The number of lives at risk, and of men to be recruited and trained over a long series of years would be reduced, without reducing the manœuvring or the offensive or defensive power of vessels of any class in the Fleet.

Rebunkering at sea—so anxious a problem with coal—would be made easy, there being no difficulty in pumping from a store-ship to a warship in mid-ocean in ordinary weather. 300 tons an hour is quite a common rate of delivery in the discharge of a tank steamer's cargo under ordinary conditions of pumping.

The many parts of the boiler fronts and stokehold plates, now so rapidly corroded by the process of damping ashes before getting them overboard, would be preserved by the action of the oil fuel, and the same remark applies to the bunker plating, which now perishes so quickly by corrosion in way of the coal storage.

Liquid fuel, if burned in suitable furnaces with reasonable skill and experience on the part of the men in charge, is smokeless. It is easy to produce smoke with it, but this is evidence of its being forced in combustion, or of the detailed arrangements of the furnace being out of proper proportion to each other. In regard to smokelessness it is, when used under conditions customary in the merchant service, not inferior to Welsh coal, and superior to any other coal ordinarily in use.

The cost of fuel in the East is less than that of Welsh coal when the cost of transport and Suez Canal dues are added to the original price of the coal as delivered in a Welsh port.

The evaporative duty required from the boilers of destroyers is greater than that required for boilers of any other type, and, whilst it is possible to burn enough liquid fuel to produce the required duty in boilers hitherto using coal at natural draught, or even coal at moderate forced draught, difficulty has been found in burning enough oil fuel in boilers of the destroyer type to produce the same duty as that realised under coal at great air pressure. The question of economy of fuel in destroyers when at full power is of comparatively little importance, but the production of the maximum power is essential; and further experiments now in process will probably solve the difficulty as it has been solved in locomotive practice on the Great Eastern Railway by the skill and enterprise of Mr. Holden.

Messrs. Yarrow have obtained some highly encouraging results in two torpedo boats built by them for the Dutch Government. Their process is to obtain the maximum speed with coal under all usual conditions of forced draught, and then to inject liquid fuel into the furnace above the coal, thus securing additional boiler duty while leaving the whole of the grate surface available for coal combustion. The result was to increase the maximum speed of the vessel by over one knot per hour. Messrs. Thornycroft have recently made experiments, and have obtained the high evaporative duty of 18·95 lbs. of water per pound of oil fuel.

The extra rapidity of raising steam with liquid fuel is undoubted, and this is a feature upon which much stress has been laid in recent Naval debates. So far as experience has shown, there is no deterioration of liquid fuel, however long it is stored before use.

Comparative Advantages and Disadvantages for Mercantile Ships.

The conditions which fuel most fulfil in the case of a mercantile vessel differ only in some respects from the conditions applying to a war vessel, the chief difference being that of cost. The question of direct cost is of little importance in a war vessel, provided the advantages above-named are really secured in practice, but in a commercial vessel the direct cost of fuel (although here also necessarily merged in other questions) is of the first importance. Whilst the supply of liquid fuel was confined to the eastern hemisphere, it seemed hopeless to expect its successful commercial competition with coal west of the Suez Canal, because of the light freight and transport charges, although for the converse reason it seemed easy for oil to beat coal east of the Suez Canal, and for some time Suez appeared to be the nearest economical station for British ships. The discovery of oil in Texas, however, and the splendid enterprise and foresight of Sir Marcus Samuel in organising a fleet of vessels for its transport and tanks for its storage have changed the situation, and there appears no reason to doubt that ports in the western hemisphere can be supplied with oil from Texas at a cost not proportionately exceeding that of coal.

The saving in stokers is considerable, although the complement should not be reduced below that necessary to assist the ship's engineers in overhauling, or in case of emergency. In some instances, a stokers' and trimmers' crew of thirty-two is now represented by a firemen's crew of eight hands, whose duty, however, is mainly cleaning and helping the engineers with their greasing. The Hamburg American S.S. *Ferdinand Laeisz* was recently reported to have discharged her stokers at Singapore in consequence of successful experience in using liquid fuel. Some of the great Atlantic liners experience a difficulty in maintaining their full-power speed as compared with the performances when the ships were new; this is not due to any deterioration in the boilers or machinery or quality of the coal, but to a difficulty in getting the trimming and stoking done to the fullest capacity of the boilers, and in the case of these vessels, whilst the saving of stokers' wages and provisions and the substitution of additional passenger accommodation would be great commercial gain, the advantage of increasing the power and speed and rendering them more uniform would be of even greater importance.

In the case of medium-sized vessels working with natural draught, the advantages of uniform steaming, irrespective of wind and ventilation, and of large reserve of steaming power, are the same for liquid fuel as for every other system of forced draught.

The greatest commercial gain, however, is the increase of weight and space available for freight. Adopting the proportion of 3 tons of coal as equal to 2 tons of oil fuel * we find a gain in weight of, say, 1,000 tons in the freight of a first-class Atlantic steamer, and a gain of nearly the whole of the bunker space, which, except for non-stowage in the hot parts, would be available for measurement freight. Allowing for these, and assuming the storage of the whole of the fuel in the double bottom and peaks, there would be a gain approaching 100,000 cubic feet of measurement made available for freight in such a vessel. The gain from substituting the new fuel in vessels of less steam power in proportion to the size of vessel would be correspondingly reduced, but it may be fairly estimated for most ships that 25 per cent. of the space now occupied by coal bunker storage could be utilised for cargo by the transfer of the fuel in a liquid form to the double bottom and other parts not now of any direct use.

The cleanliness of oiling instead of coaling passenger ships, and the saving of detention at ports of call are obvious.

The provision of storage and pumping and ventilation arrangements and of the furnace gear are a disadvantage both as regards cost and weight, and in some ships trouble and expense has arisen from boiler leakage consequent upon the presence of water in the oil and the lack of experience of the engineers, but these difficulties are now disappearing, and latest developments are in the direction of simplicity and less first cost.

Oil fires do not require cleaning, thus avoiding a prolific source of lost speed in ordinary voyage routine.

Oil Storage on Board Ship.

As already indicated, parts of the vessel hitherto useless except for water ballast or fresh water are most suitable for oil bunkers, both because of the freeing of other

* In locomotive boilers on the Great Eastern Railway, economical results of two to one have been obtained. The other advantages of liquid fuel make the value to consumers in cost of fuel still more in favour of oil than these figures.

spaces for cargo, and because of the immediate adaptability of the water ballast structure for holding liquid fuel. Lloyd's Register have published a set of rules applicable to existing vessels, which, although containing some requirements which may not ultimately prove to be necessary, are yet not unreasonable in regard to a system that had, at the time the rules were issued, all the risk of novelty. Indeed, the acknowledgments of shipowners are due to the Registry Committee and its officers for the enlightened and progressive attitude they have adopted towards this new departure in British vessels. The same acknowledgments are due to the Marine Department of the Board of Trade, whose officers have recently been closely investigating the question in detail, and who are understood to have determined that, with such safeguards as experience has shown to be necessery, there is no obstacle to the use of liquid fuel on passenger vessels. The penetrating qualities of refined petroleum, or lamp oil, are well known, and such oil will find its way through a riveted and caulked joint quite tight against water under pressure. This percolating quality does not, however, apply to fuel oil, the greater viscidity of the latter making it at least as easy to hold as water. It is believed that a ballast tank top or shell plating joint in way of oil storage, which is sufficiently riveted for water ballast, may safely be regarded as sufficient for liquid fuel, subject to testing to a head of pressure at least equal to the load water-line, and subject to all the usual requirements of separate oil-pumping pipes and connections, drainage wells, dunnage, and sounding pipes.

Some additional watertight subdivision in the double bottom is necessary in large vessels to provide against the scend of a half-empty oil tank, but in vessels of medium and small size it is believed the usual subdivision is enough.

Fig. 1 (Plate III.) shows the storage of liquid fuel on board S.S. *Murex*. This vessel is a tank steamer with all her tanks adapted as cargo holds to carry general cargo in the alternative with refined oil, and her original construction did not contemplate liquid fuel. She was built by Messrs. Wm. Gray & Co., of West Hartlepool, and about two years ago she was placed in the hands of the Wallsend Slipway Company for conversion to liquid fuel burning. Her cargo tanks have no double bottom below them, as they extend to the skin of the ship, but there is a double bottom A and B below the engines and boilers, and cofferdams C and D at the fore and after ends of the cargo space, the propelling machinery being aft. These spaces, along with the fore and after peaks E and F, have been utilised for storing the liquid fuel. The settling or service tanks G and H have been built into the vessel at the height of the 'tween decks, and the process of receiving, storing, and feeding the oil to the furnace mouths is as follows:—The flange J is coupled up to the store pipe, the oil passes along the receiving pipes K, K, K, K, and, by means of controlling valves, is directed into the fuel tanks A, B, C, D, E and F. From the fuel tanks it is lifted by pumps, and delivered to the settling tanks G and H. Each of these tanks is fitted with a steam coil for the purpose of slightly raising the temperature of the oil and facilitating the separation of any water the tank may contain from its previous use for water ballast. Each settling tank is proportioned to contain twelve hours' supply for the whole of the furnaces, and, after twelve hours allowed for settling, the water is drawn off the lower part of the tank, where it settles by reason of its greater specific gravity, and the connections opened with the furnaces, whilst, for the next succeeding twelve hours, the other tanks perform similar functions of separating the oil from water.

The drainage wells required to prevent any accidental leakage getting into the ordinary bilge are shown at M M, with separate suction pipes and delivery pipes.

Fig. 2 (Plate III.) indicates the general arrangement of storage of liquid fuel in an ordinary cargo boat with the propelling machinery amidships.

Furnace Arrangements.

These differ considerably, and it is certain that further advances towards perfection both as regards economy and maximum effect will yet be made. As already explained, the fuel is too stiff to be burned except after disintegration, and this is effected either by pulverising with steam; or by injecting the fuel under pressure, so that it breaks itself against an obstacle at the mouth of the furnace; or by vaporising by heat before the furnace mouth is reached. Mr. Howden has a modified system by which the fuel is injected into the furnace at the same time as air under pressure, such air having been previously heated by the waste gases in the chimney. This system has been very successfully fitted to the North German Lloyd steamships *Tanglin* and *Paknam* by their builders, Messrs. Workman, Clark & Co.

All the vessels on the Caspian Sea have the direct steam pulverising system, and some of them have been so running for twenty years.

As an example of the direct steam pulverising type of furnace, Fig. 3 (Plate III.) shows the furnace gear of S.S. *Murex*, already referred to. This vessel arrived in the Thames last month, after a voyage of 11,800 miles from Singapore, *viâ* the Cape, pausing at Cape Town for four days, and the furnaces not being touched from the beginning to the end of the voyage. The consumption of coal in this vessel has averaged 25 tons. Her consumption of liquid fuel for the same indicated horse-power has averaged 16 tons. A, A, A, A, are the steam-pipes; B, B, B, B, are the oil-pipes; C, C, C, C, the burners, which are hung on swivels, D, D, D, D, so as to be adjustable in position, and to be capable of swinging back to allow the opening of the fire-doors, E, E, which are equally suitable for coal when the burner orifices F, F, F, F, are closed by the pivoted slides. The brickwork is shown in Fig. 3A (Plate III.) at N, N, these being the pillars and arches against which the flame impinges in the first instance, and at K, K, K, the secondary bridges intended to baffle the flame, and prevent its too direct impingement upon the stay nuts at the backs of the combustion chambers, the tube-ends, the saddle-plate seams, and other parts liable to injury by the intense local heat generated by liquid fuel under steam impact.

Fig. 4 (Plate III.) shows the steam burner of the Rusden Eeles type in section. It will be seen that the annular orifices, both for the admission of steam and oil, are adjustable whilst the apparatus is at work, and this exact adjustment appears to be an essential condition to obtain the best results.

Fig. 5 (Plate IV.) shows the steam burner of the Holden type, which has the same facilities for adjustment, but the steam in this case passes through perforations in the outer ring A, A, and draws an additional supply of air into the furnace.

[Specimens of these burners, and also of a simple non-adjustable burner made of gas piping, were laid on the table at the Meeting.]

During the development of the liquid fuel system in the Mercantile Marine some doubt has been expressed by shipowners as to the possibility of quick reversion to coal if the supply of oil should fail, or if the cost of oil should suddenly be raised. To meet this difficulty, and to demonstrate the facility of interchange between the two fuels, the furnace of s.s. *Trocas* (Fig. 6, Plate IV.) was designed and fitted by the Wallsend Slipway Company, who have already fitted over fifty steamers to burn liquid fuel. In this case the fire-bars are left in place and covered with a layer, about 8 in. mean thickness, of broken firebrick A; behind is the bridge B, and above that again is a firebrick protection C, as already described, and a brick lining at the back of the combustion box D.

To change from oil fuel to coal, it is necessary to swing back the burners quite clear of the fire doors, and lay them back against the boiler fronts; to open the fire doors and rake out all the broken firebrick; to lay the bars with coal, and to ignite it either in the ordinary way or by means of the oil burners. These operations were demonstrated on the *Trocas* at a sea trial in September last, and in twenty-eight minutes after steaming full speed under oil the vessel was steaming full speed under coal. On the whole, however, experience of long passages is against leaving the fire-bars in the furnaces whilst working under oil, and the change from one fuel to the other can at the worst be effected in a few hours.

Fig. 7 (Plate IV.) shows the general arrangement of S.S. *Trocas*.

The consumption of steam for pulverising is estimated at 0·2 lb. per I.H.P. per hour. Large interchangeable evaporators are fitted as part of the liquid fuel outfit, usually in three interchangeable sections, and when these evaporators have been worked steadily and successfully, no ill effect whatever has shown itself upon the boilers. Two small burners in each furnace are preferable to one large burner, as being more easily adjustable, and maintaining continuity of flame without interruption. The loud whirring noise which made the earlier steam burners so objectionable has been quite abolished in those of recent manufacture.

The system fitted to the Hamburg-American S.S. *Ferdinand C. Laeisz* (Fig. 8, Plate IV.) is that known as the Korting system, and does not involve the use of direct steam for pulverising the fuel. After water separation in the manner already described, the oil is raised by a steam pump, and is heated to about 60° Centigrade by a heater on the suction pipe; it is also filtered before reaching the pump valve. It then passes through the pump, and is delivered to another heater, which raises its temperature to 90° Centigrade, and at this temperature, after further filtration, under a pressure of 30 lbs. per square inch, it is injected into the furnaces. In the act of injection a spiral, or centrifugal, motion is given to the oil by a screwed needle, and this, added to the pressure from the pump, and the softening effect upon the oil of heating to high temperature, causes the oil to spray and ignite. The fire-bars are left out, the furnaces are bricked right round, and air is admitted to the furnaces through perforated gratings, fitted with adjustable covers. This system has been so successful that the Hamburg-American Company have fitted four vessels with it.

The Meyer system, fitted in the vessels of the Dutch Steam Packet Company, is also worked without steam jet, and depends largely upon the assistance to combustion of air heated previously to its meeting the liquid fuel. Figs. 9 to 12 (Plate IV.), show a furnace on the Meyer system. The furnace is prolonged by an annular extension B. This extension is provided with spiral partitions, C, C, C, intended to give a whirling action to the air as it enters the furnace, the air as it passes through these spiral partitions being heated from the combustion inside the extension of the furnace proper. The injectors or burners for the oil are on the Korting system already described. There are dampers, E, E, E, to control the admission of air. The Meyer system has now been in use for two years on several Dutch steamers, and with great success.

In the Roumanian service, vessels have been running with liquid fuel arrangements of a generally similar character.

It is believed that the present stage of advancement of this interesting question has now been generally described, and that the widespread attention and steady application of scientific research now being brought to bear upon so important a subject must lead to great developments in the near future.

DISCUSSION.

The President: Gentlemen, we have heard a most interesting paper from Sir Fortescue Flannery upon a matter of great importance to the ships of the future, and I thought I might make a small addition to his statement about the future of oil fuel. It is this: I have just come from Egypt, and I dare say you remember a little while ago a report, which was said to be unfounded, about the finding of a petroleum field in Egypt upon the borders of the Red Sea. I inquired about it when I was there, and I was told that, although it is not yet an absolutely established fact, it will very shortly be so. They have got so far in their boring that a very large jet of gas, which scientists assert means the presence of petroleum, came from below, and they are confident, from the indications on the surface, that there is a large field of petroleum there. If that is so, it will no doubt be a source of great wealth to the Egyptian Government, and of great convenience to vessels going through the Canal, which require a supply of that fuel.

Mr. Francis Elgar, LL.D., F.R.S. (Vice-President): My Lord and Gentlemen, I am really not properly able to open the discussion, but I feel that we are much indebted to Sir Fortescue Flannery for coming here to read this interesting paper; and, although I cannot say much upon it, I should be pleased if he would allow me to ask one or two questions, in the hope of gaining still further information. Sir Fortescue Flannery has presented the oil fuel question to us, as it seems to me, in a very clear and interesting way. It is a question that is rapidly forcing itself to the front, and one that all who have to do with ships may soon require to take into practical account. On the other side of the Atlantic, owing to the recent discoveries (referred to in the paper) of oil in large quantities in California and Texas, there is a great deal now doing in the way of building ships for the utilisation of oil fuel, and of altering for oil fuel ships that are already built or building. Sir Fortescue Flannery does not state—and I do not know myself—whether there are ample means in existence, or being provided, for laying down oil fuel in this country, so as to put English shipowners in the same position as American shipowners for obtaining oil fuel at a paying rate for ships trading to and from home ports. I mean, at a rate at which oil fuel would be able to compete advantageously with coal. Another question I would like to ask is with reference to the figures upon page 55, which give the relative efficiencies of coal and oil fuel. The only kind of oil fuel I happen to know anything about, or to have any particulars of, is the Texan. Some American shipowners were going into the question of the use of that fuel a few months ago, and I was told by them that, according to the best information they had, they did not consider it was safe to rely upon a higher efficiency for this oil than would be given by allowing four barrels, or 200 gallons, for one ton of coal. Now, that works out in stowage space at about 32 cubic feet of Texan oil for 1 ton, or, say, 45 or 46 cubic feet of coal. With regard to weight, it appears to work out at about 5 tons of oil for 6 tons of coal. If my information is accurate (and I am unable to judge of that), the relative efficiencies given in the paper would be considerably altered. Of course, this is entirely a question of data, which experience alone can furnish. It would add greatly to the value of the paper if the author could give us some practical data, or actual results of experiments, that would enable a reliable estimate to be made of the quantities of different kinds of oil that are found to be respectively the equivalents, in practice, of one ton of coal.

Mr. A. F. Yarrow (Vice-President): My Lord and Gentlemen, Sir Fortescue Flannery has asked a question with reference to three first-class torpedo boats built by us for the Dutch Government and which were fitted for oil fuel. They were not fitted with the oil fuel arrangements in order to get an additional maximum speed. The case was this: the engines were proportioned, together with the boilers, to provide as much steam with coal as the engines could use. Therefore, any increased evaporation would not be of much use, because the engines were not designed for it; but the oil fuel arrangement was fitted so that, after a long run, when the fires were dirty, it might happen that the maximum speed was required, and the steam generating power of the coal could then be supplemented by the oil. It appeared to work exceedingly well, and, if I remember rightly, after running some long time, when the fires were in a dirty state and the speed fell about two knots in consequence, by turning on the oil we could regain the two knots lost. For a short time, it is, I think, quite legitimate to use oil in this way, and to consume the steam for the sake of burning the oil. Of course, without very extensive means of producing fresh water, it would not be wise to use steam continuously for burning oil, but for a short period of an hour or so, for example, it would be quite legitimate to do so, and I think the oil fuel arrangement, viewed in that light, may be considered as a very suitable fitting to any vessel at the present time. The only question I would like to ask Sir Fortescue Flannery is this: If I understand rightly, the Korting system requires no steam, and the Meyer system also requires no steam. Now my impression has been that the reason that oil fuel has not been used up to now, is on account of the difficulty that is experienced in burning the oil, except by means of a steam jet, and the consumption of water is consequently very considerable—I think the author puts it down at 0·2 per cent.

Sir Fortescue Flannery: 0·2 per cent. per I.H.P.

Mr. Yarrow: That involves a considerable loss of steam, and if the two arrangements or either of them—the Meyer or the Korting system—can burn the oil equally as efficiently as with steam, it looks as if the steam system is undesirable. I should like, however, to know whether these two methods are equally as efficient as the steam arrangement, and, if they are, why should not either of them be used and steam be abandoned? I think that if any method that does not involve the loss of fresh water could be adopted, and it proved to be a success, it would facilitate very much the introduction of oil fuel.

Admiral the Hon. Sir Edmund Fremantle, G.C.B., C.M.G. (Associate): My Lord Glasgow and Gentlemen, it appears to me that the question of liquid fuel is one which must interest us very largely in the Navy. Every executive officer and every engineer will, of course, feel that it is a very important question, and we are accordingly very much obliged to Sir Fortescue Flannery for bringing it before us. It is evidently an engineer's and a constructor's question, but chiefly a constructor's, on account of the amount of weight and space which can be saved. We all regret not to see Sir William White here, for I could imagine his delight if he were told that, in a ship like the *Majestic*, he could use oil fuel. As far as I can recollect, from what the author has said, it would be a matter of something like 600 tons which would be saved, that is, if 2 tons of oil will do the work of 3 tons of coal. If she stowed about 1,800 tons of coal, I estimate that there would be a saving of at least 600 tons in weight of fuel alone, and in addition to that, we have, as was remarked by the author, the saving of weight in the reduction in the number of stokers and all their requirements. It is, like so many other things, a question of cost, though not entirely so, because it is a question of where the fuel (whether coal or oil) is produced. Admiral Selwyn's name was mentioned, and I have often heard him, at the United Service Institution and elsewhere, advocate the claims of liquid fuel. He always used to say that liquid fuel was the right thing to use, and he always maintained that it was to be had in England, in Suffolk and in Kent—and if you bored deep enough, you would get to the shale, and it could be used for liquid fuel. On that point, I should like the author to be kind enough to tell us whether it is thought at all probable that we could find oil in this country, because, from a national point of view, it is obviously very important that we should have the oil in our own hands, and not in those of our possible enemies. It so happens that we have coal in this country, and that it is cheap; but we have no oil, and what oil we can get here is dear, and, under those circumstances, you cannot expect the Government to be very keen about fitting ships so as to burn oil. Another thing has been explained to some extent by the author, and that is this: It has always appeared to me very strange that, whereas ships in the Caspian Sea have undoubtedly for the last quarter of a century been using liquid fuel, the latter has never got further than the Caspian. That has been to some extent explained by the facilities there are for disintegrating the oil, and on account of the presence of fresh water. I do not quite understand the process which has to be carried out to make liquid fuel available (although I believe it was fully explained in the paper), but it seems to me to be a little complicated, and after hearing Mr. Yarrow, I think it is rather more so! There is one thing which everybody has remarked: We are told again and again in lectures that this fuel can be burned so as to be as smokeless as Welsh coal, and yet in our experience and in our experiments, the common expression is "that horrible ship is attempting to burn petroleum." The *Surly* was a well-known case in point. All of us have our prejudices, and we are very conservative in the Navy; and when we see a vessel smoking very badly—which is a drawback both from a military and from an æsthetic point of view—we naturally object to it. In conclusion, I hope Sir Fortescue Flannery will kindly explain to us whether there is any likelihood of getting this oil from any district in England or from other parts of the Empire; I understand that there are oil wells in Canada, but I do not think they have been alluded to by the author.

Mr. B. MARTELL (Vice-President): My Lord, this question of oil fuel has occupied so much attention, and is now coming to the front so rapidly, that I think we are greatly indebted to Sir Fortescue Flannery for the manner in which he has brought these facts before us, giving us the results of his experience, and also for the great enterprise of Sir Marcus Samuel in doing what he has to utilise oil, because, although we know that there is an illimitable supply of oil in the Empire, the fact is that we could not get at it for useful purposes. The enterprise that Sir Marcus Samuel displayed was to purchase land producing an unlimited supply of oil, and to build ships suitable for the purpose of carrying it. Now it was thought, when Sir Marcus Samuel advocated the carrying of oil—such offensive, abominable stuff as oil—out from this country and bringing home grain, which was liable to deterioration from the actual smell alone, that he was displaying considerable temerity in undertaking it. The general opinion was that this could not be safely done, that underwriters would not insure such ships, and that it was impossible to effect the two purposes. It has, however, been done over and over again, and underwriters now are just as free and just as willing to insure these ships one way as the other. I think that is a remarkable thing. We ought to give praise where praise is due, and we certainly ought to give it to one who brought forward an idea which has worked out so successfully. With reference to what Mr. Yarrow has referred to in particular, namely, the use of steam, and also with regard to the other burners that we have here on the table, Lloyd's Register Committee took a very great interest in this matter on its first introduction. There were two great points that were considered. One was the danger of having a combustible substance in the engine-room near the fires, and the second was the great waste in using steam and the difficulty in utilising it to its full extent. Well, gentlemen, it is very pleasing to hear from this paper that we now have something that will avoid the use of steam. I agree with Mr. Yarrow that, if we can find that, then the proper course to pursue is to adopt what is best, what is most economical, and what, I am sure, will commend itself to all those who have to do with this subject. The advantages derived from the use of oil on board men-of-war cannot be doubted after what is stated here. It is a matter that would place us in a very different position indeed from our present one, when we have to have either coaling transports or coaling stations for our men-of-war. Now this seems to point to a way of doing it in a manner that must commend itself to everyone who wishes to see a saving in the working expenses of our men-of-war. The Admiralty are very conservative, as Admiral Fremantle has said, and we know it to our cost. But it is very much to their credit that they have now placed ships in a position to give accurate information with regard to the difference in the expense of using oil and using coal. There is also the desirability of ascertaining accurately whether it is possible for us to look forward to obtaining oil on this coast, referred to by Dr. Elgar and Admiral Fremantle. That question is of very serious importance. I do not think we should get enough shale oil for the purpose. However, probably Sir Fortescue Flannery can tell us more about it. I well remember looking at the ship built by Messrs. Gray & Co., that he refers to here, and, although she was not, as he says, built expressly for carrying oil, still she was built with perfect cargo tanks with double bottoms, and all the arrangements of the best constructed merchant vessels, and it is very gratifying to find that she has performed her work so well that she is suitable for any trade. That only goes to show that any ship that will do for general trade purposes will do for carrying oil, either separately with grain or merely as deadweight alone. In conclusion, I can only say that we have been exceedingly gratified to listen to this paper, and I think it will lead to much good in setting the minds, more particularly the younger minds, of engineers and of those belonging to this Institution, to work. We older ones are done, we have got past that stage, and we are now looking forward to the younger ones to turn their attention to new projects of this kind, so that the old country may reap the benefit of it.

Mr. H. F. Swan (Member of Council): My Lord and Gentlemen, after what Mr. Martell has just said, I feel some diffidence in getting up to speak, as I am afraid I am not one of the younger ones to whom he has referred. It seems to me something like thirty years that we have been in the habit of sending vessels out to the Caspian fitted for burning oil fuel; in fact, the burning of oil in the Caspian was quite common before the tank steamer was in operation there. As most of you know, the composition of Russian oil is such that there is a very large proportion of "residuum," which is technically called "astatki." There was an immense supply of fuel, so much so that at times it was allowed to run into the sea, and the difficulty was, in fact, how to get rid of it at all. Hence, for a great many years, while every steamer was burning it, there was no effort made to do so economically. The burners in use were of a crude description, and often consisted merely of concentric tubes, one with oil, the other with steam by which the oil was sprayed. In recent years, when the supply of fuel in the western and eastern portions of the globe became accessible, engineers set to work to devise more useful means of burning it, and they have now arrived at a considerable degree of success in that respect. In constructing boilers for burning liquid fuel, there is a great deal to be kept in view. You must have long tubes that are small in diameter, and very large combustion chambers, otherwise the risk of damaging the boiler is considerable. I believe, in many cases, boilers have been damaged by the very means that were adopted to protect them, viz., a large amount of fire-brick. I do not quite agree with what Sir Fortescue Flannery has said regarding the water in the oil coming over from the ballast water, where oil has been carried in the ballast tanks. Some of the oil which has been used in recent years has contained a very large amount of water, as it comes from the earth, and the result has been to cause very great damage to the boilers from racking strains, owing to the burners being constantly extinguished by the water getting into them. I am a thorough advocate of oil fuel, and I am quite persuaded that it is merely a question of £ s. d. In these last few years we have seen very great developments both in the East and now in Texas, which go to prove that oil fuel in the future will be one of the elements to be taken into consideration. The comparison with the cost of coal is the only question. If the result which the author has mentioned, of the *Murex* having a consumption of 16 tons of oil as compared with 25 tons of coal, can be obtained in regular practice, then I think a very great step has been attained towards its general adoption. In speaking of the advantages of oil, especially in the case of the Navy, I do not think the author has really taken sufficient credit for the advantages gained as regards trimming, especially in war vessels, where there are all kinds of irregular spaces—six inches, a foot, or even eighteen inches wide—which cannot be used for coal storage, but into which oil would flow automatically when loading, and from which it could be withdrawn without the slightest trouble, just as easily as if it were in a square box; and that is, of course, of the very greatest importance in a war vessel.

Dr. P. Dvorkovitz (Visitor): My Lord and Gentlemen, I am a stranger here, but I have followed Sir Fortescue Flannery's paper with regard to the adoption of liquid fuel with very great interest. I have been connected with this subject in Russia, at Baku, from the beginning, and there is no doubt that the use of liquid fuel has already proved everywhere a great success from every point of view. During my last visit to Texas I personally investigated the oil fields there, and I had an opportunity of inspecting the boilers of the *Lucania*, and I made some calculations which I think will prove that an enormous saving in labour, and also in money and speed, will accrue from the adoption of liquid fuel. There are a few points in Sir Fortescue Flannery's paper, and the first is this: he regrets that there is only one place in the British Empire where liquid fuel is available, namely, Burmah. I think he should not leave out Canada, where we know there are 10,000 wells. It is quite true that the production per well is very small, but their cost is also very small. In Canada alone they produce now about thirty million gallons of oil per annum, and, now that British capital is about to develop the Canadian oil fields, I am quite sure that we have a very large source of supply there. Mr. Yarrow mentioned the question of steam. Certainly, with regard to torpedo boats, it may be a difficulty, but in large steamers, with extra condensers and so on, I do not think this question of using steam will stand in the way of adopting liquid fuel. As the author has shown, the result, based on two years' data, has been that the *Murex*, steaming with liquid fuel (taking into consideration that extra water had to be evaporated for the liquid fuel), consumed 16 tons of oil as against 25 tons of coal, and that seems to me to be a result which no doubt will be achieved everywhere. Then there is a question about water. I think it was mentioned here twice that water was mixed with the oil. Water should not be in the oil at all. The residuum in Russia, and also in Texas, is a product which in the course of the distillation of the oil—for separating out certain most valuable parts, such as refined oil and lubricating oil—is subjected to a very high temperature, in fact, to something like 400 degrees Fahrenheit, and at that temperature no water could be present. If water is present in the oil, the consumer should demand to know why it is there, because it has no business to be there. The question, therefore, is entirely, as Colonel Swan has said, one of £ s. d. It is quite true that, until lately, this question could not be solved, that is, until the discovery of the Texan oil fields, but this discovery has shown that oil could be delivered at 7s. 6d. f.o.b., and, taking the cost of freight and storage accommodation here in the United Kingdom and everywhere else, we ought to be able to receive it at from 30s. to 32s. 6d., so that it would be available at every port in the United Kingdom; and I maintain that, at such a cost as that, liquid fuel might be substituted for coal in a great many cases. I would just tell you what I have told my American friends, that they should not go too far. They should never think that oil will be a substitute entirely for coal everywhere. After all, the production of coal is now 800 million tons all over the world and the production of oil is only 20 millions, and it is a far cry to say that we are going to substitute 20 million tons of oil for 800 million tons of coal. Under certain conditions, with passenger vessels, men-of-war, and for various other purposes, I foresee there is a very good market for liquid fuel, but it will not disturb in any way the market for coal.

Herr L. GÜMBEL (Associate): My Lord, I would like to ask Sir Fortescue Flannery if he would be kind enough to give us the evaporating power of Mr. Thornycroft's boiler when using coal, because I consider those figures to be rather important for comparison.

Sir FORTESCUE FLANNERY: Mr. Thornycroft, who is here, would be better able to do that.

Mr. J. I. THORNYCROFT, LL.D., F.R.S. (Vice-President): Burning oil is not so simple as it would appear to be at first sight, and I cannot say that we have, up to now, perfected the thing. The evaporation which we obtained seems to be very high, but before sending the record to the Institution I should like to verify it.* Perhaps there is one thing which I might state, and that is, that the amount of water used in some of our experiments certainly was a great drawback. We made experiments some years ago, because we were asked to fit liquid fuel apparatus in a lifeboat, and for that purpose we found that there was very considerable difficulty. With regard to burning oil without using steam to disintegrate it, I think what has been said in the paper is very interesting and valuable, and, if I might give it an explanation, I would like to say that, if you take the oil and disintegrate it with steam, you do two things: you heat the oil with the steam, and you do a considerable amount of work in what some people call "pulverising" it (although I hardly like to use that word when I am speaking of a liquid). If you are going to do it more purely mechanically, and not to use steam—if you attempt to use cold air, the expansion of the air cools it, and the surface tension of the liquid is greater. In Sir Fortescue Flannery's description here, the oil is heated and filtered. If it is heated to a high temperature, the more it is heated the less the surface tension of the liquid becomes, and the less work there is required to reduce it to the proper fineness for combustion. By that method, I certainly see that there might be success, which, I understand from the paper, has already been attained. Getting over the difficulty of using steam, as was pointed out by Mr. Yarrow, is a very great advantage.

Mr. J. MELROSE, R.N. (Visitor): My Lord and Gentlemen, as a humble representative of H.M.'s Navy, I should like to make a few comments on the very interesting paper which Sir Fortescue Flannery has read to us. It has been presented to us in a very attractive aspect—persuasively attractive—so much so, that if I were not an engineer in H.M.'s Navy, I should think the time had come to resort to the use of liquid fuel immediately without any hesitation. But we have only had presented to us the advantages of its use, and that under very roseate conditions. It would be as well, perhaps, to inquire into some of the possible disadvantages; and reference has been made by one or two of the gentlemen who have spoken, to the possible disadvantages of having water intermixed with the oil. The effect of that, as the author has pointed out, has been to injure the boiler when oil has been used. The amount of water that may be present with the oil is a very variable quantity. I have heard on very good authority, from a gentleman who has had some experience with the composition of oil fuel, that he has found it to vary from 3 per cent. to 40 per cent. It is manifest that if we have to use a fuel that can contain 40 per cent. of water, some of its properties must be very considerably depreciated, because

* Mr. Thornycroft regrets that the duration of the trials referred to was not long enough to allow him to publish the apparently very high results recorded.—ED.

an admixture of water with oil not only produces the effect described in the paper, but it extinguishes the burners during their operation, and the result of that is, or may be, that, if the burner is not immediately re-ignited, and the oil still continues to flow as well as the water, the high temperature existing in the furnace would be presently succeeded by an explosion resulting from the vaporisation of the oil that was admitted subsequently to the extinction of the burner, and the admixture of the air coming into the furnace at the time. That may be a danger. Mr. Martell, I think, referring to the Admiralty, charged them with being very conservative, and doubtless they are so in a great many cases, but the Admiralty have had this under their notice for a considerable number of years now. Twenty-five or thirty years back, they were experimenting with liquid fuel at the time the late Admiral Selwyn advocated its adoption, and many experiments have been carried out at Portsmouth. The result of those experiments showed that the additional value of liquid fuel as compared with coal did not reach anything like the 50 per cent. claimed in the paper, but reached possibly 33 per cent.—one-third. The theoretical calorific value of the best Welsh coal may be taken as 15 lbs. of water evaporated per pound of coal. The theoretical calorific value of oil fuel of the chemical constituents given in the paper would work out to 20 lbs. That shows an advance of one-third, and not one-half. The value of the oil theoretically would, then, be one-third more than the value of the coal—that is, in its evaporating power. I have been engaged for some little time past in making experiments with oil fuel, and I hope to prosecute them still further, and to obtain a considerable advance upon what has already been arrived at, which, I am sorry to say, does not reach anything like the figures given in the paper. It is quite possible that, although the difference between the theoretical calorific value of liquid fuel and coal is not so great as is represented in the paper, it by no means follows that in actual practice such as has come under the observation of Sir Fortescue Flannery in the Mercantile Marine, a value of 50 per cent. better than that of coal could not be attained, because if the liquid fuel is effectively and perfectly consumed during combustion, there would be no loss as compared with coal. There must be a residue from coal—I mean apart from the actual inorganic substances—that cannot be burned. There is always ash which would still have some calorific value if it were burned out, and there is always loss in good coal going up from a stokehold plate with the ashes pulled out from the fires. There is no loss of that kind with oil. The use of oil in the merchant service has undoubtedly been attended with a considerable measure of success. It may be doubted, perhaps, whether the full power has ever been realised by the use of oil fuel alone. I think it has been approximated to, but possibly not exactly realised without the production of smoke. The production of smoke is a very easy matter, as is pointed out in the paper, with unskilled attendants. Smoke, when it does come, comes very plentifully, as Admiral Fremantle has pointed out with reference to the *Surly*. It comes without any doubt, and so black that you cannot see the sun through it on a bright day. This means that a great deal of the oil that is going into the furnace has not been consumed, but is lost entirely, and a great deal of it goes up the funnel. In the merchant service, happily, their boiler arrangements are sufficiently large and roomy to enable a rate of combustion to be effected which will pretty nearly realise their full power, but, unfortunately, in comparing the conditions which would exist in a Navy boiler and a Merchant service one, we find them so enormously different that the two problems are entirely distinct, and the solution must be attained by some other means, at present unknown. One of the dominant factors in the case of the combustion of liquid fuel is undoubtedly space—the space available for combustion. The practice in the successful steamers in the merchant service has been to realise pretty nearly their full power at a combustion of about 2½ lbs. of oil, per cubic foot of space—that is, of space available for combustion. In a boiler of the *Surly* class, to which reference has been made by Admiral Fremantle, to attain the required power, we should have to burn, at the very least, 12 lbs. per cubic foot of space with perfect combustion, but with imperfect combustion we should require to burn from 15 lbs. to 20 lbs. The very large difference between the combustion of $2\frac{1}{2}$ lbs. per cubic foot and 20 lbs. is one that introduces inconceivable difficulties. For effective combustion of course, it is necessary to have introduced with the oil the required quantity of air, but, so far as my experience goes, it is quite uncertain and unknown what is the best place to admit the necessary quantity of air. The difficulty connected with air admission has not been unknown in the merchant service, because we have had some recorded cases of the re-ignition of the gases in the root of the funnel, producing overheating in the uptakes. We have had the same thing under observation in an experimental boiler that we have had at work recently, and it is very curious to observe the effects of a difference in the position of the entrances for the admission of air. In the paper, reference is made to the furnace, and the author points out that it is necessary to have a special arrangement of furnace to ensure combustion of the oil. With a sufficiently large furnace, and a proper amount of brickwork, I do not think there would be any difficulty whatever attending the combustion of the oil—a combustion that should reach very near perfection. There would be no smoke, no residue, and no trouble other than that due to the presence of water to some extent in the oil; but, to secure the best method of consuming the oil in large quantities in a furnace, it would be absolutely essential to have brickwork for the maintenance of the necessary amount of heat to provide for ignition and subsequent combustion. That condition of furnace can be provided for in a merchant service boiler, but not so readily in a Navy boiler, mainly owing to the want of space. Another of the attractive features attending the use of oil given in the paper is that we are going to abolish the stoker. Now, the stoker is a very useful man in our service. The term "stoker" is not a good one nowadays, for the Naval stoker really has to be a mechanic, and a fairly good one, too; but, at any rate, I do not think it follows, because we can get efficient burners, or get oil to burn in sufficient quantity to reach our power, that we are going to get rid of the stoker altogether. Sir Fortescue Flannery hardly appreciates the difference between steaming a merchant vessel and steaming a man-of-war. With regard to the former, as soon as she is clear of the port, her speed is uniform until she makes her next port of destination, and the stoker, as he turns on the oil, and if the burners are satisfactory, can practically sit down and watch his gauge glass and have nothing whatever to do. But with a man-of-war, especially in fleets, as everyone knows, the variation in the speed is so continual that it would not be an exaggeration to say that an alteration of speed would probably take place on an average every three minutes, and very often less than that. An alteration of speed means the alteration of the burners, because the rate of supply of the oil to the fire, and also the rate of supply of the steam, must be controlled as frequently and as rapidly as the change of speed, so that you always will have to preserve skilled labour sufficient to be able to control the burners, even when they are perfected, and I do not think we have quite reached that stage yet. I did not quite understand from the paper what the author's object was in showing us the more primitive type of burner, whether he wished us to infer that there was very little difference in the efficiency of the burner, between the primitive type and the more highly developed types of Holden's and Rusden & Eeles' burners. The most efficient type of burner would be that which is capable of most perfectly diffusing or spraying the oil under either compressed air or steam, and which most finely atomises it and prepares it most readily for its conversion into gas by the application of heat.

Sir Fortescue Flannery, M.P. (Member): My Lord and Gentlemen, let me claim the indulgence of the meeting whilst I refer in the order in which the observations were made to most, if not all, of the speeches which we have heard during this exceedingly interesting discussion. Dr. Elgar referred to the alteration of vessels that is taking place in New York. I happen to have some particular knowledge of what is going on in New York, and many of the plans of the alterations have been before me. I can assure Dr. Elgar that the statements which have come from some of the oil people in Texas, so far from confirming his ideas of 4 to 3 instead of 3 to 2 are so highly coloured——

Dr. Elgar: They are not my ideas; it is what I have been led to understand from the other side.

Sir Fortescue Flannery: The ideas that were communicated to my friend.

Dr. Elgar: 200 American gallons are equal to 160 English gallons.

Sir Fortescue Flannery: I follow that, and I am obliged. I was going to say that the figures which reached me from America from some of the oil people there are entirely different from the figures which have reached Dr. Elgar. It has been claimed that the Texas oil is equal to 2 to 1 as regards coal. I have regarded that as an American statement, and I have not put it forward at all. I should like to answer Dr. Elgar's point in a word, and to say that whilst I know Borneo oil from actual experience from log books which have come before me, from voyage to voyage, containing results extending over two years and lesser periods as regards burning oil, I have no actual experience with Texas oil as yet to show the relative calorific value in practice of that oil. The Texas oil people claim that it is better than the Baku oil. The chemical analysis seems to indicate that it may probably be taken to be as good, but whether in practice it will give us such satisfactory results as the Baku oil, I am not able to say. Then Dr. Elgar asked as to the oil stations, and the regularity of the supply of the oil to this country. Of course, nothing could be more important than that question, and nothing is more actually applicable to the immediate question which shipowners have to deal with. I may say that there is already in London a station which has a storage capacity of 30,000 tons of liquid fuel, and that regularly, almost every week now, vessels come there to receive into their bunkers liquid fuel from the tanks which are situated at Thames Haven. Tanks for liquid fuel are in greater or less progress towards completion in the United Kingdom at Liverpool, at Southampton—where a site has been arranged for—at Birkenhead, and at Barrow; and on the Continent the Hamburg-American Company are erecting or have partially erected their own storage tanks in Hamburg for their own fleet. Now, my Lord, as to the question of the actual comparative efficiency of coal and oil fuel. I have referred in my paper to the *Murex*, giving the figures of 25 tons of coal on an average over a period of six years or more, and the figures of 16 tons of oil with approximately the same I.H.P. and speed of vessel. I was asked whether there were not other examples which I might quote. I am very glad therefore to mention a larger vessel, the *Strombus*, built by our friend Colonel Swan, who has addressed us in the course of this discussion, and who is perhaps more entitled to speak on this subject than most other shipbuilders, because he is chairman of the Wallsend Slipway Company, who have fitted some fifty vessels for liquid fuel. The *Strombus* burned about 43 tons of coal during her career and about 30 tons of oil—it has been as low as 28, but I always prefer to keep well within bounds, and therefore I speak of it as 30 as compared with 43 tons of coal. Those are practical figures. There are also two other vessels, the *Haliotis* and *Trigona*, built about four years ago by Messrs. Armstrong, Whitworth & Co. from Colonel Swan's designs. The figures as regards coal on the outward voyage, and the oil which they use now and have used for fully three years, are rather more favourable in point of consumption than those which I have just given. Then there is the *Ferdinand C. Laeisz*, the first steamer of the Hamburg-American Company which gives about the same ratio of results, that is to say, about 3 to 2; it is rather less favourable in the case of the *Ferdinand C. Laeisz* than with regard to the other vessels which I have named. Mr. Yarrow mentioned in his short speech a most important point, that is, the doing away with the consumption of steam for spraying the oil. No engineer requires to be told how injurious it is to boilers to take away the live steam and so reduce the fresh water; unless, of course, you have the power, by evaporators, of supplying the water again, not only fresh but also hot, otherwise your boilers are sure to suffer injury. We find it necessary to have about double the evaporative power that the makers tell us the evaporators will produce. I do not want to say a word against evaporators, but in practice, allowing for inefficiency in management and so forth, we find we cannot very easily have less than double the evaporative power which is estimated. I have had figures taken out very carefully, with the result I have given of about 0·2 I.H.P. per hour for the consumption of steam. But although that is not a very large consumption, and although the total economical consumption, which has been named, includes the raising of steam for the purpose of pulverising, yet no one can deny that it is of the utmost importance to get rid of the steam if you possibly can. Therefore, I join with Mr. Yarrow in looking hopefully to these other systems of vaporisation and of injecting the oil under pressure; but I am bound to say that, up to the present time, they have not been so economical, in point of consumption and of general economy of oil consumed *versus* power produced, as the steam-driven apparatus. There is not a very great difference, but the difference, such as it is, is rather against the pressure-driven apparatus as compared with the steam-driven apparatus. As I said in the paper, however, we are all learning on this subject, and I believe that a very great improvement will ultimately arise in the direction of Howden's system, in the regenerating system as regards the air, as well as the other systems which I have named. I may say there are two sister ships building at this time—very large vessels—intended to carry liquid fuel; and it is proposed to have in one of them a steam-driven apparatus and in the other a pressure-driven apparatus, with a view to getting the very best comparative results on a careful system of scientific comparison. Admiral Fremantle referred to the question—and this is really a most important and almost a political one—of the supply, and he mentioned the shale which is known to exist in large quantities in Scotland and elsewhere. I have reason to know that this matter is not at all overlooked; in fact, after the admirable speech of the Chief Inspector of Machinery, who was officially told off to deal with the question, we can see that it has not been overlooked at the Admiralty. The question of obtaining supplies within British territory is of the greatest importance when the Admiralty deals with this matter. There is not only the necessity of obtaining a sufficiently economical result on board ship, but also of being certain that if you adapt His Majesty's ships to work, at all events from time to time, upon liquid fuel, you will have a supply of that fuel insured without complications in time of war. I believe that geological investigations into the large quantities of shale, not only in Scotland but on the east coast of England, would be very well worth while. I sincerely hope the Admiralty will follow on with those investigations, because, although there are deposits of oil in Burmah, and in Canada, yet they are often too remote even there to rely on under all circumstances. Mr. Martell referred to the question of petroleum and the carrying of cargo. That has only incidentally to do with the subject that we are discussing to-day, but I may mention that this system has been so economical as regards cargoes of oil, succeeded by cargoes of rice and other such perishable goods in the same hold, that underwriters are now preferring to even make a slight difference of premiums in favour of tank vessels to carry rice and grain from the East, the main reason being that there is much subdivision and unusual ventilation in these tanks. The practical effect is that there is no damage whatever to the cargo, and that underwriters find fewer claims in regard to that. My Lord, the real difficulty in this question as applied to vessels of war is the one pointed out by one of the previous speakers, and that is, as to getting the full amount of work out of the limited space and weight of boilers that can be allowed in war vessels. The boiler space and weight for a mercantile vessel are practically unlimited—I mean in a cargo boat—and there is really no difficulty in getting as much duty out of those boilers as is required, or as would be given by coal under ordinary circumstances; but in the Navy, especially in torpedo boats, that is the great difficulty. It is one which I am certain will be overcome (if it can be overcome by anybody) under the care of Sir John Durston and Mr. Melrose. I feel that this Institution should place on record, even if it is only by my own humble voice, the appreciation which it has of the progressive policy of the Admiralty in this matter. Sir John Durston has shown himself thoroughly progressive since he came to his present position, and in no respect is he more so than with regard to this question. Lord Selborne mentioned the matter very fully in his recent Memorandum, and I feel certain that whatever can be done by way of investigation and experiment is now thoroughly taken in hand. Mr. Melrose spoke of the comparison between the evaporative efficiency theoretically of coal and of oil. He mentioned that 15 lbs. of water was the theoretical evaporative power of coal.

Mr. MELROSE: The best Welsh coal.

Sir FORTESCUE FLANNERY: Yes, the best Welsh coal; and 20 lbs. the same comparative figure as regards oil, but that there was not anything like the margin there which practice had shown of 3 to 2 in evaporative duty. My Lord, the facts are, I think, as Mr. Melrose pointed out, that oil is more effectively burned and more completely consumed in practice than coal. Whether that be the explanation or not—and I believe, with Mr. Melrose, that it is—there is no doubt that in practice these figures have been sometimes (though not always) obtained, and there seems to be very little difficulty in repeating them, especially after the engineers have had a little experience and practice in working them.

Mr. MELROSE: Pardon my interruption, Sir Fortescue Flannery, but your comparison was probably with North-country coal.

Sir FORTESCUE FLANNERY: The comparisons I have made have been with all sorts of coal.

Mr. MELROSE: There are 50 per cent. better results with the oil.

Sir FORTESCUE FLANNERY: The comparisons I have made have been with all sorts of coal; that is to say, coal that a mercantile vessel gets where she can.

Mr. MELROSE: Not the best Welsh coal.

Sir FORTESCUE FLANNERY: No, not the best Welsh coal.

Mr. MELROSE: That would explain some of the differences which appear to exist between the comparisons which I have referred to, made between the best Welsh coal and liquid fuel.

Sir FORTESCUE FLANNERY: That will explain some of the differences, but the main explanation is that the coal fuel is less effectively burned and less completely consumed than the oil fuel.

The PRESIDENT: I am sure you will all join with me in moving a hearty vote of thanks to Sir Fortescue Flannery for his most interesting paper.

The following written contributions to the discussion have been received:—

Mr. E. L. ORDE (Member): The subject of "Liquid Fuel for Ships" is one of the highest importance, and the intrinsic merits of the fuel are so great as to deserve perhaps even closer investigation than has hitherto been applied for ascertaining the most satisfactory method of burning it in boiler furnaces. To effect complete combustion of liquid fuel is, as Sir Fortescue Flannery shows in his paper, not a difficult matter in boilers working at ordinary rates of evaporation, but in the case of boilers from which a high evaporative duty is required, the problem becomes somewhat more complex, and the principal difficulty which has to be overcome is that of maintaining the furnace temperature at the high point which is required to burn the first instalment of the carbon monoxide, before the flame, by contact with the cold furnace plate or tube, becomes cooled to the point at which the vapour condenses and prevents the completion of the first stage of combustion. The difficulty has been got over, in some cases, by lining a portion of the furnace with refractory material, but in boilers from which a high duty is required, so great a sacrifice of efficient heating surface cannot be made, and some other means of attaining the same end must be adopted. The solution of the problem would appear to lie in the direction of vaporising the fuel before admitting it to the furnace, and thereby presenting it in a form in which combustion may take place at once, and, in taking place, produce the highest initial temperature possible with such a fuel. The question of air supply to the furnace, when burning large quantities of such fuel, is one that requires very careful consideration. The conditions of combustion differ from those which obtain with a solid fuel, inasmuch as the air in the latter case is, by natural or mechanical means, brought into closer contact with every particle of the fuel before it is allowed to enter the furnace. In the case of liquid fuel, however, where, as the author has explained, the combustible is admitted into the furnace in a finely divided state, it is obvious that only the outer layers can come in contact with the air, and it is not until these outer layers are burnt, that the inner layers can be brought in contact with sufficient oxygen to secure their complete combustion. When burning comparatively small quantities of oil in the furnace of Scotch boilers, this requirement can be satisfactorily complied with, but the case is somewhat different in water-tube boilers, where the furnace space is necessarily contracted in proportion to the amount of fuel that must be burnt to obtain the necessary boiler duty, and some arrangement of air compressors such as are used in connection with Belleville boilers would seem to be necessary. I raise these two points with some diffidence, in the hope that some means may be found of solving the only difficulty which seems at present to prevent the application of liquid fuel to such craft as torpedo-boat destroyers, and others fitted with high-duty boilers, for which its many great advantages would seem to make it essentially suitable.

Mr. E. T. Agius (Associate): In order to show how much interest is taken in the introduction of liquid fuel for use in marine and land boilers, I may mention that steps are being taken to erect tanks for the reception of liquid fuel, and negotiations have been entered into with several Governments in the Mediterranean to allow of their erection. At Malta, a tank has already been in existence for some years, but it has been exclusively used for the storage of burning oils. With the object, however, of supplying liquid fuel, a large tank is being erected which will meet the necessary requirements of the trade. Special interest attaches to the site where these tanks are erected, as it is a short peninsula, in the middle of the harbour, where steamers can be supplied with petroleum by means of pumps at any time of the day or night, and in any number at one and the same time. I should think that over twenty steamers could be so supplied at one and the same time.

Mr. J. M. Adam (Associate): Attention has been drawn to modes of combustion depending on the use of steam jets for pulverising the oil, and the objections thereto on grounds of economy of steam. This method is not the best for other reasons also. There is usually a deposit of moist slime which may be smokily consumed on hot brick, but which is lost to thermal efficiency. On contact with the oil, particles of steam are condensed, and each minute sphere of water is covered by a film of oil, and this compound particle is incombustible. The question raised by Mr. Melrose regarding the necessity of combustion in confined spaces is intimately connected with the supply of a combustible atmosphere, there is certainly no lack of concentration in the fuel. The required atmosphere is not supplied by steam, but is, on the contrary, displaced by it to some extent. Indeed, steam is specially out of place in the presence of hydro-carbon, where vapour of water forms so large a proportion of the products of combustion. The most satisfactory disintegrator is atmospheric air, and a pressure of 20 lbs. per square inch suffices. I am in possession of a nozzle or burner—the result of much experiment—so contrived that the air has control of the oil, and a proportionate adjustment of the elements can be made with one valve, and it seems to be free from the defect feared by Dr. Thornycroft, perhaps owing to the comparatively light pressure employed. Several small jets should be used to each furnace, each at its best adjustment, and variations of power met by shutting off or re-igniting one or more; for any attempt to vary the consumption of a jet is apt to disturb the due proportions of the elements, and results in smoky combustion or accidental extinction. Such an arrangement also minimises the danger of quenching by contained water, these extinctions being caused by isolated globules, instantaneous in passage as they are inevitable in effect. These interruptions become imperceptible amidst a group of burners.

THE NAVIPENDULAR METHOD OF EXPERIMENTS, AS APPLIED TO SOME WARSHIPS OF DIFFERENT CLASSES.

By Captain G. Russo, Naval Architect, Royal Italian Navy, Member.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 20, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.

Note.—A demonstration was given in the Library of the Society of Arts by Captain Russo with the navipendulum, which had been lent for the purpose by the kind permission of the Italian Minister of Marine.

At the Spring Meeting of 1900, I had the honour of reading before this Institution a paper on an experimental method of ascertaining the rolling of ships on waves. In a branch of naval architecture illustrated by such splendid theories and elaborate investigations as those connected with the names of Mr. W. and Mr. R. E. Froude, Mons. L. E. Bertin, Sir W. H. White, Mr. P. H. Watts, Captain Kriloff, and many others, the object of that paper was to show how the problem of rolling among waves could have a simple and expeditious solution, if transferred from the field of analytical or graphic research into the experimental field, by a process founded on the great principle of mechanical similitude.

That paper indicated the principles on which I had based my method of research, described the construction of the navipendulum, and the details of the apparatus for producing the effects of wave motion, and further showed the results of certain preliminary experiments, which were such as to confirm the reliance to be placed on the whole method.

A short time afterwards, my apparatus was temporarily erected at the Italian Ministry of Marine in Rome, where I am occupied with my official duties, and where I have had an opportunity of carrying out a great number of navipendular experiments. In response to the invitation to contribute some further results of experiments, I have collected and put in order the results obtained in comparative tests of the rolling of certain battleships; to bring these results before the Institution forms the principal object of this paper. I have, besides, the honour of showing the apparatus used in my experiments, and I am thus enabled to show in a practical way how the experiments are made.

In spite of my keen desire to submit to the examination and discussion of the Institution an apparatus, which in Italy has come into use to furnish data in the designing of ships—data which could otherwise only be obtained with the greatest difficulty—I might still perhaps not have decided to do so, had it not been for the kindly interest taken in the matter by our Vice-President, Mr. A. F. Yarrow, who, having been present at some experiments in Rome, encouraged me in every way to bring the apparatus to this meeting. I feel it, therefore, my duty to take this opportunity of thanking him, and at the same time to thank the Secretary, who has been good enough to take charge of everything connected with the reception of the apparatus, and the necessary preparation for its working. I have finally to offer special thanks to His Excellency the Italian Minister of Marine, Vice-Admiral Morin, who, having heard of the matter, kindly allowed this apparatus, destined for the use of the Royal Italian Navy, to be temporarily taken out of Italy.

I.—Notes on the Method of Making Navipendular Experiments.

I will begin by a few brief notes on the method of making navipendular experiments, for the better understanding of those of the members who care to examine the apparatus, and who have not before them my preceding paper.

In the mechanical system by which the rolling movement of ships amongst waves is reproduced on a small scale, there are two principal parts: one represents the vessel, and consists of a pendulum of a special shape, to which I have given the name "*Navipendulum*"; the other represents the waves, and consists of an apparatus capable of causing on the navipendulum the same effects which the waves, assumed to be regular and of trochoidal form, produce on the real vessel.

A navipendulum (Fig. 1), arranged to represent a given ship, is so made as to possess, on a small scale, all the geometrical and mechanical properties of the corresponding ship, which are in play in her rolling motion. The form of the line *b b*, which determines the swinging of the pendulum along a plane surface, depends on the *shape of hull* of the ship; it is a line parallel to the line of the centres of buoyancy, but passing very close to the centre of gravity. The centre of curvature M of the line *b b* on its middle point corresponds to the *metacentre*. The two weights W, W are so arranged that the centre of gravity G of the navipendulum occupies, in respect to the line *b b* and to the point M, a position corresponding to that of the *centre of gravity* of the ship. The distance between the points M, G represents, on a reduced scale, the distance between the metacentre and the centre of gravity of the ship, or in other words the *metacentric height*. The position of the weights W, W, besides being governed by the condition that the centre of gravity shall be in the desired spot, is also governed by another condition, namely, that the *moment of inertia* of the navipendulum should correspond, in

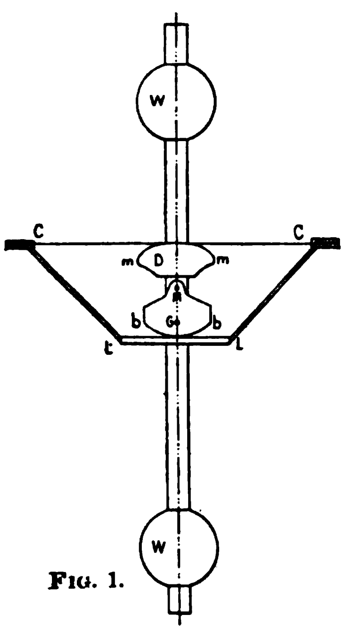


FIG. 1.

the due ratio of similitude, to that of the ship. This second condition is fulfilled when the *period of oscillation* of the navipendulum corresponds to that of the ship. To be precise, if the ratio of similitude for the linear dimensions of the ship and the navipendulum be λ, then, in order to satisfy the general law of similitude, it is necessary that the ratio of the time should be $\sqrt{\lambda}$.

Besides fulfilling the above conditions, the navipendulum, to correspond to the ship, must meet, in its oscillations, resistances corresponding to those which the ship meets in water when it oscillates ; artificial resistances must therefore be brought in, and brought in in such a manner that the loss of amplitude in each oscillation will be the same for the navipendulum as for the ship; in other words, the navipendulum must have the same *curve of extinction* as the ship. By curve of extinction I mean a curve which has for its abscissæ the numbers of successive swings, and for ordinates their amplitudes. The means adopted to obtain this consist of the addition of a block of wood D, and a leather strap C C stretched above it. The shape of the surface *m m* of the block D is determined in quite a practical way, by successive experiments, according to a process of trial and error; the tension of the belt is regulated by a stretcher and by a dynamometer. A direct verification, made before proceeding to carry out a series of experiments, by getting a curve of decreasing amplitudes, allows one to correct, if necessary, the tension of the belt, so as to obtain the exact curve desired.

As is apparent, the curve of extinction is one of the data of the problem. If we are treating of a ship already existing, it may be the result of a rolling experiment. If, on the other hand, we are investigating the properties of a vessel still at the stage of design, or if, in an existing vessel, we wish to study the effect of alterations of the period, or the effect of additional resistances, then the curve of extinction is obtained from a model and tank experiment in calm water.

Having arranged the navipendulum as described, its rolling on the plate L L, fixed horizontally, corresponds with all desirable approximation to the rolling of the vessel in still water.

The inclinations of the navipendulum are transmitted to a registering apparatus by means of a system of articulated pieces, so designed as not to affect the free oscillation of the navipendulum, and only bringing in very slight resistances.

Passing now to the wave-motion apparatus, it has its characteristic feature in the cinematic law of the movement given to the plate L L (Fig. 1, page 77), on which the navipendulum rests. Such a motion corresponds, on a reduced scale, to that of a portion of the surface of the wave, considered as trochoidal; or, in other words, corresponds to the motion assumed by a raft put to float on the wave we are considering.

In Fig. 2 a series of waves are represented, corresponding to successive positions of the same wave W W. Let us suppose that the wave is moving from left to right, and consider a limited portion of its surface, or, in other words, a little floating raft. The black lines represent the positions of the little raft at the passing of the waves. Its motion is determined solely by three conditions :—(*a*) its middle point moves at a uniform rate along a circumference, whose diameter H equals the height of the wave; (*b*) the normal to the floating raft, elevated through its centre, is constantly directed to a point P, which lies on the vertical through the centre O of rotation, at a distance from it $\overline{OP} = \frac{L}{2\pi}$, L being the length of the wave from crest to crest; (*c*) the time employed in the rotation is equal to the period T′ of the wave.

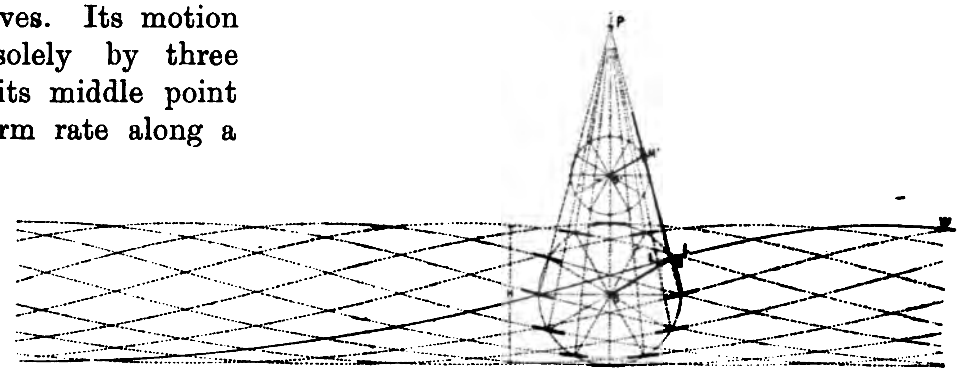


Fig. 2.

Now this little raft may be easily supposed to be freed from the action of the liquid surrounding it, and put in motion, under identical conditions, by specially designed mechanical means. It will also be easily understood that this raft, with its motion, can be represented on a small scale. Here also, as for the navipendulum, the law of mechanical similitude holds good, and, if the ratio of similitude is λ for lengths, it must be $\sqrt{\lambda}$ for time.

In the apparatus, a motion of this kind is given to the plate L L (Figs. 1 and 2, pages 77 and 79) on which the navipendulum rolls, and which constitutes the essential part of the whole mechanism. I need not repeat here the mechanical constitution of the apparatus. I will confine myself to recalling that the plate L L is supported by a rotating arm O M; the normal M P to the wave surface, or to the plate, is materially represented by a rod M M′ (Fig. 2), acting as a governing lever to regulate the inclinations of the plate. This rod, by means of a pivoting boss, situated in M′, is made to pass constantly through the extreme point M′ of a rotating arm O′ M′, which, in its uniform rotation about the point O′, keeps constantly parallel to the arm O M. A glance at the photograph (Plate V.), allows the pieces which in Fig. 2 are represented by the lines L L, O M, O′ M′, M M′, to be easily recognised.

In order to set the apparatus to represent, with a given scale of lengths λ, a given wave of the length L, height H, and period T′, the following conditions must be fulfilled :—(*a*) the length of the arm O M is to be $= \lambda \frac{H}{2}$; (*b*) the length of the arm O′ M′ is to be $= \overline{OM}\left(1 - \frac{2\pi \overline{OO'}}{\lambda L}\right)$, as may be easily verified; (*c*) the period, or the time of revolution, must be $= \sqrt{\lambda}\, T'$.

It is important to point out that in the same way as the period T′ of the wave is a function of the length L, so equally the rotatory speed of the apparatus must be regulated as a function of the lengths assigned to the arms O M, O′ M′. During the experiments, this speed must be kept steadily uniform; for this purpose a Valessie indicator has been set, insuring the strictest regulation of the motion. The keys of this instrument having been suitably arranged, according to the required number of revolutions of the apparatus, the hand of a special clock must be seen to be at rest; otherwise, it shows some motion forwards or backwards, thus indicating that the speed of the machine must be increased or decreased.

The apparatus is run by an electric motor; when motive power is supplied by a battery of cells (as it has been in all my experiments) the motion is very regular by itself. The number of revolutions may be made to vary from 10 to 77 a minute. The required speed is obtained by means of a pair of stepped cones of pulleys, and by electric resistances introduced; besides this, there is a gradual electric resistance under the control of the assistant observing the indicator, who is thus enabled to correct the small fluctuations of speed, which otherwise could take place. To insure greater regularity of the motion, balance weights are employed in connection with the three principal axles of the machine.

The recording apparatus, besides registering the oscillations of the navipendulum, registers also the oscillations of the normal to the wave with respect to the vertical. By this means one is enabled to verify, after the experiment, whether the pre-arranged speed has been exactly maintained. In the meantime, a scale of time in seconds is inscribed. The resulting diagram is of the form shown in Fig. 3.

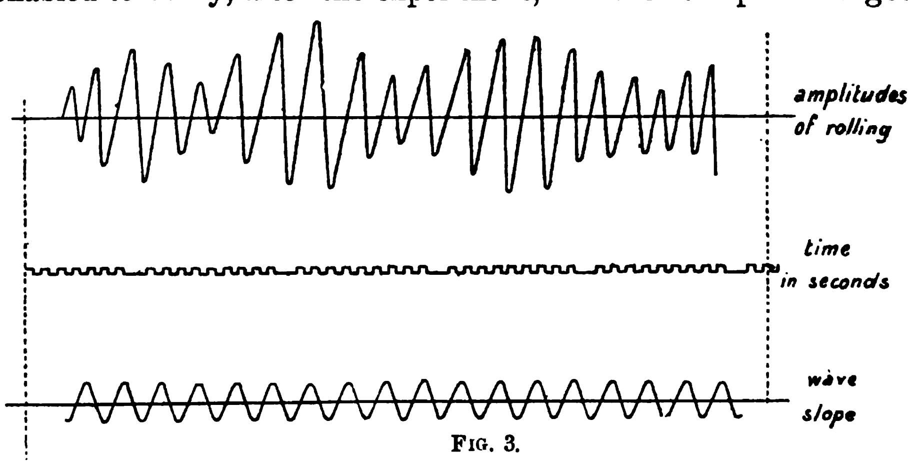


Fig. 3.

The recording apparatus is supported by some parts of the machine, which make it accompany the rolling plate in its rotation. The motion of the recording apparatus only differs from the motion of the plate in that it remains always upright; thus the registered angles of inclination, both of the navipendulum and of the normal to the wave, are referred to the vertical.

In proceeding to experiment, the navipendulum is arranged to represent a given ship on a given scale, and the apparatus is so disposed as to represent, in the identical scale, waves of given lengths and heights. For a certain time after the apparatus is started, and until the right speed is reached, we have a period of irregular motion during which the navipendulum cannot be abandoned without a danger of displacing itself or sliding along the sustaining plate, owing to the fact that its own weight and the centrifugal force do not give a resultant at right angles with the said plate. When the right speed is reached, which is shown by the Valessie indicator, the *apparent weight* of the navipendulum, as well as of any other object placed at the middle point of the plate, is directed perpendicularly to this; from that moment the navipendulum can be abandoned to free oscillation without the danger of any sliding. In the same way, the ship floating on a swell has no tendency to slide along the slope of the wave. A round body put on the plate of the apparatus remains steady, no matter how great the slope of the plate may be, thus demonstrating in an evident manner the most characteristic property of the wave motion.

The navipendulum may be abandoned to free oscillation, upright or inclined, with or without an initial rotatory speed. For reasons that will be explained hereafter, I follow the practice of leaving the navipendulum stationary and upright on the crest of the wave. Then a series of oscillations begins, with amplitudes successively increasing and decreasing at nearly equal intervals of time; the maximum amplitudes reached go on successively decreasing, and the minimum amplitudes increasing, until the differences disappear altogether, and the navipendulum falls into the uniform rolling known under the name of *forced oscillation.*

During the forced rolling any alteration in the regular motion of the apparatus or of the navipendulum, such as would be due to a momentary variation in the speed, or a light shock against the navipendulum, is enough to alter the regularity of the rolling, and to originate a reproduction of the phases of maximum and minimum amplitudes, followed, after some time, by the return to the conditions of forced rolling.

The mechanical similitude between the motion of the plate and that of the water in wave motion may be demonstrated by means of a direct experiment, which brings out clearly the peculiar properties of the trochoidal wave. We connect with the plate a glass containing water, with its brim parallel to the surface of the plate. When the apparatus is in motion, if we cause the level of the water to rise up to the brim, the water does not overflow, no matter how great the slope of the represented wave may be. Anyone observing this experiment may imagine the liquid to be a small part of a real wave, the glass itself forming part of the wave, and having the same specific gravity as the water, so as to be exactly supported inside the liquid, and one may imagine a real swell to pass continually before his eyes with a period equal to that of the revolution of the apparatus. Such a supposition explains the reason for which the level of the water follows, without any overflowing, all the inclinations alternately given to the glass to the right and left; and, so far, the well-defined movement, depending on the connection of the glass with the apparatus, offers a demonstration of the laws to which the masses of water constituting a real wave are subjected.

Other demonstrative experiments could be made with navipendulums to show practically, for instance, the effects of synchronism between ship and wave, the different behaviour on the same wave of two ships, one being very stiff and stable, the other having a very small stability.

For instructive purposes, experiments of this kind could be shown in schools, employing apparatus much smaller and more simplified in respect to the actual mechanism, which has been constructed for a different object, namely, for obtaining exact measurements concerning the various classes of ships, and thus getting comparative data about their degree of steadiness.

II.—Some Considerations on the Principal Characteristics of the Curves of Oscillation and on the Commencement of Rolling.

It is well known that on a continuous and regular swell a vessel can roll in an infinite variety of different ways, according to the particular angle and angular speed, with which we may suppose her to start at any particular instant; but, whatever may be the particular conditions of rolling under which the motion starts, if the swell keeps perfectly regular, and if outside influences do not intervene, the vessel, after having rolled in a non-uniform way for a time of more or less duration, ultimately falls into the uniform forced oscillation in which she keeps perfect time with the waves.

This phenomenon is naturally reproduced in the navipendular experiments, and constitutes one of the most visible characteristics of the curves of oscillation obtained from the apparatus. Only in the case of the navipendulum being abandoned to the action of the apparatus, with an angle of inclination and an angular velocity well determined, and dependent on the precise instant of the commencement of motion during the rotation of the apparatus, would it be possible to fall into forced rolling without passing previously through a period of irregular rolling, more or less continued according to the conditions at starting and to the resistance to which the ship is exposed.

Now the study of non-uniform rolling, that precedes forced rolling, is of the greatest importance, as Mr. R. E. Froude has taught us, when, from the analytical or graphic results, or from the experimental results, it is desired to extract useful data comparable to those which occur in practice.

"The ideal assumed uniformity of swell does not exist in practice. And hence, in proportion as, on the one hand, the resistance is small, and any initial non-uniformity of rolling dies hard (so to speak), and, on the other, the non-synchronism is considerable, and the alternations of amplitude rapid, the continual fresh disturbances due to the non-uniformity of swell, avail effectively to maintain non-uniformity of rolling, in spite of the tendency of resistance to eradicate it. Thus a tendency to continued non-uniformity of rolling is in practice a characteristic feature, and an important one, of rolling in a non-synchronous swell."

I have copied in the last paragraph words that are used in Mr. Froude's paper,* read before this Institution in 1896, as nothing could better express the reasons why the rolling of ships, under ordinary conditions, is far from that regularity that is characteristic of forced rolling.

I ought here to say, that, in interpreting the characters of analogy which the various curves of oscillation present, and in the reasoning concerning the way of defining the criterion amplitude, I have been guided by the great principle established by Froude in the above paper, that, although the non-uniform rolling of a given ship on a given swell can assume different aspects one from the other, yet it always consists of the appropriate forced oscillation, with or without a free oscillation superposed. I am also inclined to say, that only after having carried out many experiments on rolling with navipendulums, is one able to appreciate all the importance of that and other principles laid down in Mr. Froude's paper of 1896.

It is necessary for us, then, to study the characteristics which the non-uniform rolling of a ship on a regular swell presents; and in order to do this, the best means is to refer to the curves of oscillation, where all the elements coming into play in the phenomenon are graphically represented.

In order to study a concrete case, let us take for example a curve of oscillation A (Fig. 1, Plate VI.) obtained from a navipendular experiment in which the navipendulum has been initially abandoned to the action of the apparatus, with a casual inclination and angular velocity, in an arbitrary instant of the revolution of the apparatus. Abscissa values of the curve A correspond to time; ordinate values, with regard to the base line X, represent the inclinations of the navipendulum with respect to the vertical. The

* "The Non-Uniform Rolling of Ships," by R. E. Froude, Esq., F.R.S.: Trans. I.N.A., Vol. XXXVII. (1896), p. 306.

sinuous bottom line represents, in the same scale of time and of angles, according to which is traced the curve A, the inclinations of the normal to the wave, from instant to instant, referred to the vertical.

From an examination of the curve A, it is deduced :—(*a*) That the non-uniform rolling tends gradually to become a uniform rolling, in which the oscillations have the same period as the wave; (*b*) that the angles of roll pass alternately through maximum and minimum values at regular enough intervals of time; (*c*) that the maximum values go on continually decreasing, until they become reduced to the amplitude constant for forced rolling; (*d*) that the minimum values decrease to a certain point with laws similar to those of the decrease of the maxima, then gradually increase, until they fall into the constant amplitude for forced rolling; (*e*) that there is a certain phase of the motion, in which oscillations are reached of minimum amplitude very near zero.

On the line of rolling some auxiliary lines can be traced, which serve to explain more clearly the peculiarities of the phenomenon, namely, a sinuous line B, passing through the upper vertices of the rolling curve, and a curve B′, passing through the lower vertices; a curve C, passing through points of maximum amplitude of the curve B, and one C′ passing through points of maximum amplitude of the curve B′; two curves D and D′, passing through points of minimum amplitude of curves B, B′. Finally, we can trace two axes Y, Y′, parallel to the axis X, and representing the prolongations of the straight lines in which the lines B, B′, C, C′, D, D′, finally merge.

It follows from the theory of the superposition of the two component oscillations, and it is confirmed by the experiments :—

(*a*) That the curves B, B′ are symmetrical with respect to the axis X, and consequently also the curves C, C′ and D, D′ are respectively symmetrical with respect to the said axis X.

(*b*) That the curves D, D′ are the same as the curves C, C′, simply displaced in a perpendicular direction with regard to the axis X, through an interval equal to the distance of the axes Y, Y′; that, in consequence, the curves C, D′ are symmetrical with respect to the axis Y, and the curves D, C′ are symmetrical with respect to the axis Y′.

(*c*) That, commencing the rolling in different ways, always in the same swell, the curve A, in general, is changed, as are also the curves B, B′; but the lines C, C′, D, D′, and the axes Y, Y′ remain unaltered. There are, however, an infinite number of curves of rolling, for which the curves B, B′ also remain unaltered, this being analogous to what takes place in free oscillation in calm water, where we can have, according to the angle with which it sets out, an infinite number of curves of oscillation, the curve passing through points of maximum amplitude remaining always unaltered, that is, the curve of declining angles remaining unaltered. And, more exactly, turning to the theory of superposition, if one supposes the curve of declining angles to remain unchanged, the curve of free oscillation to be altered, and the curve of forced oscillation to be displaced with reference to it in such a manner as to leave unchanged the points in which the coincidence of phases takes place, then the condition is verified in which the curves B, B′ undergo no alteration.

According to the theory of the superposition of two component oscillations, the curves C, C′, D, D′, should be merely the curve of declining angles in still water oscillation: this is, only in an approximate way, verified in the experiments; in fact, sometimes it is observed that the non-uniform rolling lasts for a longer period than the free oscillation on the still plate. But if we remember that every alteration, however small, in the regularity of the wave motion, has the immediate effect of introducing non-uniformity in the rolling, we must allow that the divergency is to be attributed to those small disturbances which inevitably accompany the experiment, however high a degree of precision the apparatus may possess.

Again, theory teaches us that the interval of time from one undulation to another of the curve B should be n T, where n, taken always positive, $= \pm \frac{T'}{T - T'}$, T being the period of the ship, and T′ the period of the swell. This is confirmed by the experiments. Only it is to be noted that the theory allows T to be constant, and hence also n constant, for great or small angles of rolling, whereas in the experiments the co-efficient n feels the influence of the variation undergone by T according as the oscillations are great or small.*

The characteristics of which we have spoken up till now, of the curve of oscillation and the auxiliary curves which may be traced from it, are clearly visible in the example represented in Fig. 1, Plate VI.; and, indeed, we must add that they are also clearly visible in the greater number of practical applications of the experimental method, that is, in the case of ships and waves in conditions not outside the ordinary limits. But it is important to notice that, even when this is not the case, that is, when we consider ships of excessively short or excessively long periods, or with special disposition of weights, form of hull, &c., and waves of unusual period, dimensions, and inclinations, the curves of oscillation may indeed assume aspects at first sight irregular; but, on examining them attentively, one recognises that they present no contradiction to the laws which we have indicated above with regard to the curves of rolling in general.

We may cite, as examples, the curves represented in Plate VII., which, with their

* *See* foot-note, p. 94.

apparent anomalies, only give greater confirmation to what we have established concerning the characters of curves of oscillation.

It was necessary to examine the said characters, in order to be able to decide with confidence what method should be adopted, in carrying out the experiments, with regard to the initial conditions of the rolling; and also to be able to establish in a concrete manner the significance of those elements, which are deduced from the curves of oscillation, and which may be considered as the ultimate result of the experiments.

The curve of oscillation has in itself neither an initial nor a terminal point. Hence the necessity for establishing some convention, with the object of rendering comparable the effects which different conditions of sea may produce on a given ship, or those which a given condition of sea may produce on different ships.

One might, for example, arrange matters in such a way that, after the ship has acquired the forced rolling, a disturbing influence should intervene to alter the uniformity of the rolling, and establish the magnitude of this disturbing influence as a function of the wave, or of the ship, or of both together; and hence consider as a *criterion amplitude* the maximum angle which the ship can assume after the instant in which the disturbing influence has exercised its action.

Although this idea may seem rather vague, there is a simple way of understanding its application. If, in fact, we grant that the rolling begins in a given position of the ship with respect to the slope of the wave, with a given inclination and a given angular velocity, and these three elements are not in the same proportion to one another as they would be in the case of forced rolling, this is equivalent to the introduction into the forced rolling of an instantaneous disturbance, greater or smaller according as the pre-arranged conditions diverge more or less from those which should have been in the latter movement.

According to the method of Mr. R. E. Froude, founded on "the common experience that, when rolling is far from uniform, the most violent rolls are generally those which shortly follow a moment of approximate quiescence," it is supposed, in the first place, that the motion begins with no roll. It is the simplest and most acceptable of the conventions that can be made, but it needs to be more exactly determined, for it is possible to begin *with no roll* in as many different ways as there are points, in the variable slope of the wave, in which the ship may be supposed initially stationary and upright. In the same way the navipendulum may be abandoned, stationary and upright, to the action of the apparatus in the infinite number of positions which the plane of support assumes during a complete revolution.

Among these infinite ways there are two which deserve particular attention :—

(*a*) The first is when the ship is supposed stationary and upright in that same instant in which, following the forced rolling, it would be at its greatest inclination, which is equivalent to abandoning the ship, upright, to the action of the swell at mid-slope of the wave.

(*b*) The other way is when the ship is abandoned, upright, to the action of the swell, when the normal to the wave is vertical, that is, on the crest, or in the trough of the wave.

Mr. R. E. Froude has adopted, as a complement to the proposed convention, the solution (*a*); in fact, he starts with opposition of phase between the two component oscillations, at the moment when the free oscillation gives an amplitude equal to that of the forced oscillation, so that the net amplitude is *nil;* but in a note to his paper he recalls that Mr. W. Froude, in his classic theory published in 1861, dealing with the unresisted rolling, has adopted hypothesis (*b*); in the same note are indicated the differences which arise according as we take one or the other condition for the beginning of rolling.

After a mature examination of the question, I decided to adhere, in my experiments, to the second solution, that is, to establish as the initial point of the rolling the instant at which the ship is on the crest of the wave, supposing that at that instant the ship is upright and animated by an angular velocity $= 0$. A similar condition is easily realised in navipendular experiments by holding the navipendulum still until the opportune moment, and then suddenly releasing it.*

This solution, in my case, has the advantage of greater facility over the other, which would have required, before each experiment, a preliminary determination of the exact position corresponding to the angle of maximum forced rolling. But besides this, in my opinion, it possesses some advantages, for it enables one to give to the *criterion amplitude* a definition which, although still conventional, is more comprehensive and also more simple, as it does not involve the conception of the two component rolls.

As I have shown above, either of the solutions is equivalent to introducing an instantaneous disturbing cause into the forced rolling. This disturbing cause is reduced, in substance, to a certain quantity of rotatory energy added to or removed from the ship, instantaneously, during its forced rolling.

*** An improvement, which will be introduced into the apparatus, consists in adding a piece of mechanism for holding the navipendulum still, and releasing it automatically at the right moment.**

We know that the forced rolling is analogous to the unresisted rolling of the ship in still water, only supposing that the action of the couple of stability is augmented or diminished in such a proportion as to modify the period of oscillation, changing it from T, proper to the ship, to T′, proper to the swell. Now in rolling without resistance, the rotatory energy of the ship remains constant. The same may be said of the ship on the wave in forced rolling. It is certain that for each oscillation the ship loses as much energy to the resistance as it gains from the impulse due to the passing of the wave; but since the experiments do not show appreciable differences in the characters of the curves of rolling in calm water or in forced rolling, we may admit that for the whole duration of the forced rolling the total rotatory energy of the ship remains unchanged, and exactly equal to that constant quantity of energy which the ship would possess if it were to roll, without resistance, through a constant amplitude equal to that of the forced rolling, under the action of a couple varying, with respect to the couple of stability, so as to allow for the varying period.

To bring the ship to a vertical position at the instant at which it is at its greatest inclination (case *a*) is equivalent to subtracting from it a quantity of rotatory energy equal to that which would be necessary, *vice versâ*, to bring it from the vertical position to the angle of greatest inclination. If the vertical were the instantaneous position of equilibrium, which is not the case, the ship would lose instantly all the energy it possesses, and would be abandoned, inert, to the action of the sea from that instant.

Similarly, to cause the ship to be upright and stationary instantaneously at the moment when it is on the crest or in the trough of the wave (case *b*), is equivalent to subtracting from it a quantity of energy equal to what would be required to bring it from this latter position to the inclination and angular velocity which exist, in the forced rolling, at the passing of the crest or the trough of the wave. But since the vertical (in case *b*) represents the instantaneous position of equilibrium, the ship, stationary and upright at that moment, is in fact abandoned inert to the action of the swell with a quantity of initial rotatory energy = 0.

This is the principal reason why the second solution (*b*) seems to me preferable to the first (*a*). Following this method, and taking as *criterion amplitude* the maximum angle to which the action of the swell brings (or might bring) the ship, this criterion amplitude may be simply defined as the maximum angle to which the swell can bring the ship, when the latter is abandoned to the former absolutely inert, without rotatory energy of its own, either actual or potential.

It may be observed that the starting-point chosen represents a particular case among all those in which the ship may be supposed initially inert and without rotatory energy; in fact, such a state of things may be verified for all the points of the slope of the wave, so long as the ship is supposed stationary and conveniently inclined so as to be in its position of equilibrium. But the positions of equilibrium cannot be easily determined,* and hence it is difficult to ascertain experimentally if in all cases the criterion amplitude is identical. Yet, from a quantity of facts observed in the navipendular experiments, it seems to result that this is effectively the case.

Turning now to Fig. 1 (Plate VI.) it will be observed that the curves D, D′ meet in a certain point P. Corresponding to this point, the amplitudes of the two component oscillations are equal to one another; if we suppose that they are here in opposite phases, the conditions proposed by Mr. R. E. Froude are realised. That particular curve, of the type of the curve B, which passes through the point P, gives us, in its first maximum ordinate to the right of P, the criterion amplitude according to the hypothesis; this turns out certainly less than double the maximum angle of the forced rolling.

Without entering into minute explanations about some curious peculiarities of the curve B, explanations which it would be difficult to give in sufficiently clear form, I shall limit myself to indicating that the starting point of the rolling, when the solution which I have followed is adopted, is to the left of the point P if the period T of the ship is greater than the period T′ of the swell; it is to the right in the converse case. In the first case, which occurs most frequently, there results a maximum amplitude superior to that caused by starting at the point P, and such as to surpass, in some cases, double the angle of forced rolling; and this in a greater degree as T is greater with respect to T′, as may be gathered from the diagrams of rolling relative to different classes of ships, which form the principal object of this paper. In the second case the maximum amplitude arrived at is less than that caused by starting from the point P.

In conditions approximating to synchronism the two methods lead to results which do not differ sensibly from one another.

In the same Plate VI. (Fig. 2) is seen a curve of oscillation obtained with the same navipendulum and in the same swell from which was obtained the curve Fig. 1. The form of the dotted line passing through the points of maximum inclination, and constituting the *curve of rolling*, suggests the following observation :—

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* The procedure indicated by Mons. L. E. Bertin is summed up in his paper "Position d'Equilibre des Navires sur la Houle" (Mémoires de la Société nationale des Sciences naturelles et mathématiques de Cherbourg, T. XXXI., 1898).

In Appendix III. to my preceding paper, "An Experimental Method of Ascertaining the Rolling of Ships on Waves" (Trans. I.N.A., Vol. XLII. page 48), I have indicated a procedure from which a method may be deduced for determining the positions of equilibrium of ships on waves.

If we must judge the rotatory energy which the ship possesses during the roll, from the amplitude of the angles which it describes, and from the maximum angular velocity which it acquires in each roll, we may consider the dotted curve as indicating the fluctuations which are verified in the quantity of rotatory energy possessed by the ship. Starting from zero energy, the ship gradually gains energy, thanks to the impulses which it receives from a certain number of successive waves; it reaches a maximum, after which the waves have the effect of subtracting energy, and thus reduce it to a certain minimum value; then the ship begins again to acquire energy, and acquires it up to a certain maximum, which, however, is inferior to the maximum previously reached, and so on. The dotted line shows that the rotatory energy possessed by the ship fluctuates about a certain mean quantity of energy equal to that which becomes permanent in the definite forced rolling, when the curve of rolling becomes a straight line parallel to the base line; the fluctuations gradually decrease in intensity, and finally disappear altogether.

An exhaustive investigation of the question does not demonstrate that this line represents an exact diagram of the quantity of energy; nevertheless, it confirms the idea that we may regard it as a line approximately indicating the variations of the fluctuation in the rotatory energy of the ship.

III.—Comparative Results of Navipendular Experiments concerning Different Ships.

The navipendular experiments, the results of which are summarised on Plates VIII. and IX., relate to the Royal Italian battleships *Re Umberto* and *Regina Margherita*, each of them representing a class of ships; and also to H.M.S. *Revenge*, a representative of the *Royal Sovereign* class. Before taking into consideration these particular results, I shall have two more remarks of a general character to make.

First of all, it is to be noticed that, notwithstanding the great number of elements which are in play in the rolling of a ship (dimensions and shape of hull, metacentric height, moment of inertia, period of oscillation, size, form and position of bilge keels, &c.), yet all these elements are implicitly represented in two diagrams, which are sufficient to give all the necessary *data* for proceeding to navipendular experiments concerning any given ship. These are the diagram of statical stability (curve of righting moments) and the diagram of declining angles, the latter being accompanied by the indication of the natural period of the ship. In such curves, having peculiar characters for every ship, all the elements are combined, by which the ship acquires, so to speak, a certain special physiognomy in her behaviour on waves. The combination of the two diagrams into a single one—the diagram of rolling—which would answer all the questions about rolling in a seaway, may be considered as the *desideratum* to aim at.

Secondly, I must point out that, even if limited to represent for the different swells only the amplitudes of maximum oscillations (the criterion amplitudes) and the amplitudes of the forced oscillation, a diagram of rolling would prove much too complicated, had it to include all the possible heights which may be considered for any wave of a given length. As long as the practical purpose of the experiments is to ascertain the comparative rolling of different ships, we may content ourselves with establishing a certain series of waves of increasing lengths and heights practically proportional to those observed in ordinary real waves, and then adopting the same series of waves for the different ships to be compared. This is the way I have proceeded in my experiments. I have considered waves of increasing periods from 8 to 19 seconds, their heights corresponding to their lengths, in the arrangement of the apparatus, according to the formula $H = \frac{L}{60} + 1^{\text{met.}}$. It may be seen that for short waves, as, for instance, that of 8 seconds (length 328 ft.), the height considered (8·75 ft., viz., about $\frac{1}{37}$ L) would be too small; but the fact should be taken into account, as stated in my preceding paper, that when small waves are dealt with, the apparatus must be arranged to represent a fictitious wave height, somewhat smaller than the wave height actually considered. I have not made inquiries, thinking it of no practical importance, as to the precise wave height to which the trial height corresponds; I think it may suffice to know that for the smaller waves among those considered in the experiments, the trial wave slope is somewhat below the corresponding wave slope in the real case, whereas for the larger waves the difference is practically *nil*. Besides, it may seem superfluous that such large waves as those reaching the period of 19 seconds should have been subjected to experiments; but I did so because, although such large waves are very rarely met with, yet the case is frequent of waves having a great apparent length and period relatively to the ship, according to the speed and direction of advance of the ship.

A few words will suffice for an explanation of Plates VIII., IX. and X.

Plate VIII., *Re Umberto.*—Fig. 1 gives the diagram of statical stability under ordinary load condition. From this diagram Fig. 2 has been deduced, representing the metacentric evolute, and also the rolled curve (parallel to the curve of centres of buoyancy) according to which the blades of the navipendulum have been worked out. The two diagrams (Figs. 1 and 2) are common to the ship and to the corresponding navipendulum, the only difference being in the scale. Figs. 3 and 4 give the diagrams of declining angles in still water in two different conditions, viz., before and after the addition of bilge keels. The numbers marked along the base line of each diagram correspond to successive swings, say from port to starboard, or *vice versâ*. The variation in the period of swing is made apparent by the distance between the dotted ordinates, in accordance with the scale of time (in minutes) as marked underneath.

The curves of declining angles were previously determined by tank and model experiments, carried out for the sister ship *Sardegna*,* supposing the same results to be available for the *Re Umberto*. Moreover, it was necessary approximately to extend these curves in the upward direction beyond the limits of the tank experiments. For these navipendular trials the ship is considered under the following conditions:—

Displacement	13,893 tons
Metacentric height (1·48 met.)	4·85 feet
Period of a single swing, for small angles...	6·90 sec.

Figs. 5 and 6 summarise the results of the experiments, viz., the maximum amplitudes of rolling on swells of different periods and the amplitudes of the forced oscillations.

PLATE VIII., *Regina Margherita*.—The experiments relating to the battleship *Regina Margherita*, now in course of construction, were carried out some months ago, before the ship was launched. As previously stated, the navipendular method may be employed even for ships still in the stage of design. By tank and model experiments for still-water oscillation, the curves of extinction have been ascertained for different values of the period, between certain limits given by calculation. The natural period for small angles, as anticipated by calculation, will be from 7·5 to 8 seconds; these two extreme conditions have been subjected to experiments, supposing the metacentric height to remain unchanged in both cases, and also the bilge keels as actually fitted; of course the extinction diagrams are different in consequence of the different period of swing.

The ship is considered under the following conditions:—

Displacement	13,490 tons
Metacentric height (1·20 met.)	3·94 feet
Period of a single swing, for small angles, ...	7·5 and 8 sec.

Figs. 11 and 12 give the results of the experiments.

PLATE IX., *Revenge*.—The curve of statical stability reproduced in Fig. 1 is taken from Sir W. H. White's paper: "The Qualities and Performances of Recent First-class Battleships," read before this Institution in 1894. From Fig. 1 the geometrical elements of the hull represented in Fig. 2 have been deduced; from these the peculiar blades for the navipendulum have been worked out. However, with reference to the conditions of load and stability, in which rolling experiments were made on the

* See G. Rota, "La Vasca per le Esperienze di Architettura navale nel R. Arsenale di Spezia," p. 86.

Revenge in October, 1894, January and February, 1895,* I have considered some approximate mean conditions, as indicated below:—

	No bilge keels.	With bilge keels.
Displacement	14,000 tons	14,000 tons
Metacentric height...	3·61 feet	3·58 feet
Period of a single swing for small angles	7·73 sec.	8·07 sec.

Here also it has been necessary to approximately extend the curves of declining angles beyond the range of the experiments. However, it is my opinion that this operation, given the character of the curves, cannot involve any great error.

Figs. 5 and 6 (Plate IX.) show the results of the experiments. In Plate IX. some curves of oscillation are also reproduced, as obtained from the experiments concerning the ship without bilge keels.

Plate X.—Coming lastly to an examination of the different curves of rolling obtained from the experiments mentioned above, many remarks could be made by comparing them to each other. The two curves of maximum amplitudes of the *Re Umberto* (Fig. 1, Plate X.) offer a clear demonstration of the known fact that the addition of bilge keels gives no great advantage on waves of moderate period, when the ship rolls through small angles; whereas the greater advantage is obtained the more nearly synchronism is reached between the ship and the swell. The same conclusion may be inferred from comparing the two curves relative to the *Revenge* (Fig. 2, Plate X.).

A comparison of the curves of rolling of *Re Umberto* and *Revenge*, both without bilge keels (Fig. 3, Plate X.) shows a curious phenomenon, viz., that the *Re Umberto*, having the shorter natural period for small angles, reaches the critical condition of synchronism on longer waves than the *Revenge* does with a longer natural period. This is a consequence of the variation of the period with the amplitude of swing. The curve of declining angles for the *Re Umberto* (Fig. 3, Plate VIII.) shows that the period of swing gradually increases with the amplitude, whereas the curve relating to the *Revenge* (Fig. 3, Plate IX.) shows that the period is nearly isochronous up to 10 or 12 degrees, after which it gradually decreases; in such a difference we find an explanation of the observed phenomenon. On the other hand, the reason of the different law of variation in the two cases is to be found in the different shape of hull, of which an idea is given by the midship sections (Fig. 5, Plate X.); in fact the metacentric evolutes of the two ships are of an apparently different character (Figs. 2, Plates VIII. and IX.). The *Re Umberto* has a great initial metacentric height, but the point M descends, and the metacentric

* Sir W. H. White's paper, "Our Battleships—Notes on Further Experiences with First-class Battleships." Transactions I.N.A., Vol. XXXVI. 1895.

height rapidly diminishes, as the ship heels; at an angle of 40 degrees the metacentric height is reduced by nearly a third of its initial value. On the contrary for the *Revenge* the point M rises, as the ship heels to a certain angle, after which it descends; at an angle of 40 degrees the metacentre is very nearly in the same position as it was initially. Through swings of a large amplitude, the couple of stability acts on the *Re Umberto* with a comparatively lower energy than on the *Revenge*, and the effect is an increase of the period for the former ship and a diminution for the latter.*

Another point deserving attention is that the addition of bilge keels has been of greater advantage to the *Revenge* than to the *Re Umberto*. Now the bilge keels of the latter ship are 165 ft. long (50 met.) and 3·28 ft. deep (1 met.); the *Revenge* has bilge keels 200 ft. long, 3 ft. deep. In my opinion, the greater efficacy of the bilge keels on the *Revenge* must be attributed to the different form of the midship section (see Fig. 5, Plate X.) rather than to their slightly greater area. In the *Revenge* the midship section allows the bilge keels to be placed at the most protuberant part of the contour, and therefore in a condition exceptionally favourable to their action. Thus the navipendular experiments give a fair confirmation of one of the statements made by Professor G. H. Bryan in his paper about the action of bilge keels.†

A glance at the diagrams of rolling of the *Re Umberto, Regina Margherita*, and *Revenge* fitted with bilge keels (Fig. 4, Plate X.) shows that, as could have been anticipated, a characteristic curve with peculiar features belongs to every ship, in the same way as every ship has its peculiar curves of stability and of declining angles.

I do not say that a diagram of rolling could be such a curve as to give *all* the elements a naval architect would like to have at his disposal in order to get information about the probable behaviour of a new ship at sea, and about her degree of steadiness; but I think that when diagrams are available concerning many ships in actual service, it will then be possible to anticipate, by comparison, the probable qualities of a newly designed ship. Even if all the features of a ship are already

* Whether the mechanical similitude between the ship and the navipendulum holds good also in that which relates to the period of swing through large angles, is a question about which I have made some investigations. Certainly the way in which the resistance is applied involves a certain disturbing couple, which adds its effect to the action of the couple of stability; but numerical determinations prove that the amount of the additional couple is trivial as compared with the magnitude of the couple of stability. No doubt, the law of variation of the period with the amplitude of swing chiefly depends on the law of variation of the couple of stability, or, in other words, on the shape of hull; now this important factor is exactly represented, in the navipendulum, in accordance with the law of mechanical similitude. Whence it may be concluded that the period of swing for large angles, in navipendular experiments, cannot diverge very much from following the same law of variation as that for the actual ship.

† Transactions I.N.A., Vol. XLII. page 200.

determined, yet recourse may be had to navipendular experiments for deciding whether, and to what extent, the addition of resistance is necessary.

Should it be needed to inquire about the efficiency of *water chambers* in rolling among waves, that will be easily attainable by actually adding water chambers to the navipendulum.

But, confining ourselves to the case of bilge keels, navipendular experiments should be regarded, it seems to me, as a very useful complement to experimental tank trials. Supposing bilge keels of various sizes and positions to be added to a ship, tank trials can tell how much speed we lose for each addition; navipendular experiments can tell how much steadiness we gain; thus we may gather the necessary elements to decide from exact experience to what extent we may consent, either to a sacrifice of speed to steadiness, or to a sacrifice of steadiness to speed.

Such a question acquires greater importance when very fast ships of new design and of unusual proportions and forms of hull are dealt with, as, for instance, is now the case with the Italian battleships of the *Vittorio Emanuele* class, for which navipendular experiments have been ordered.

The practical problem of extension and proportion of bilge keels has been one of the objects which I have had in my mind, while proceeding with my researches these last few years.

To bring these researches before the Institution has been a great honour for me, and it will prove highly beneficial to the continuation of my studies, since the discussion of the method will most probably contribute to diminish the defects of a new system of investigation, in which not a few difficulties have necessarily been met with.

DISCUSSION.

The following contribution was received from Mr. R. E. FROUDE and read by the Secretary:—
I am extremely sorry not to be able to attend the reading and discussion of Captain G. Russo's paper. I have not even found time to study the paper closely enough to be able to comment on it in any way that could be worthy of it. I must content myself with saying that, as far as I can see, I agree with every word of the paper from beginning to end. Touching the questions considered in pp. 87, 88, between solutions (*a*) and (*b*), there is opportunity indeed for a good deal of argument. It is possible there might in the end be found to be on this head some difference of view between Captain Russo and myself. But certainly, assuming that, as he says, solution (*b*) is the more convenient of the two for navipendular work, I should myself in his case adopt it, as he has done. It was for reasons of convenience only that I preferred to use solution (*a*) in my paper of 1896.

Mr. Archibald Denny (Member of Council): My Lord and Gentlemen, Mr. Froude has anticipated me in what I was going to say in congratulating Captain Russo. I should like to have the opportunity which Captain Russo has had of studying rolling by this machine. In the paragraph at the end of page 94, he does not say that the navipendulum gives all the elements a naval architect would like to have at his disposal in order to get information about the probable behaviour of a new ship at sea, but he suggests that by constantly repeated experiments and using the results obtained from actual ships, one would be able to find out the co-efficients. I think that is the correct use of this machine. With regard to the question of tank trials for bilge keels, I think I can give Captain Russo a more valuable result than that of a tank trial. I can give him a full-sized trial of the effect of bilge keels on speed. In the case of a vessel that did not quite come up to our predictions of speed for power on her first trial, it was suggested that the bilge keels were wrongly put on, so we took them off entirely. Of course, there are always errors in indicator diagrams, and also small errors in taking speeds, but we could not find any difference at all with bilge keels or without them, and I think that at sea, instead of reducing the speed, there is every reason to believe (in a ship given to rolling) that the speed will be increased by the addition of bilge keels. I will just add that I covet that machine, and I think that the Institution owes Captain Russo a very great debt of gratitude for the trouble he has taken in getting his navipendulum brought here and showing us his experiments. Every one of them has succeeded, and that is not always the case when a machine is asked to go through its paces.

Professor J. H. Biles (Member of Council): My Lord and Gentlemen, I am afraid that this is a subject that I know very little about, but I should like to take the opportunity of thanking Captain Russo for the trouble he has taken on behalf of the Institution, and in the advancement of this obscure subject, and to say, like Mr. Denny, how much I should like to have had the opportunity of carrying out a somewhat similar investigation. We are all aware that a trochoidal wave is a beautiful geometrical conception, and I suppose most of us have been content to accept the researches of the elder Froude, and leave the matter at that point. We know that if a vessel has a long period of rolling in still water, she is not likely to roll a great deal if she meets with waves that are not of so great a period as herself. We all believe that, but when we go to sea and find how a vessel rolls, we are disposed to think there is something wrong with the sea. Now I believe I am perfectly correct in saying that this is the first time we have had a complete investigation of the rolling of a ship in which the variation of period due to a varying angle of roll has been taken into account. We have usually considered ships as rolling isochronously, and now for the first time we have had taken fully into account the effect of non-isochronous rolling in a ship, that is to say, while we know, in rolling through an angle of, say, 10 degrees as compared to an angle of 5 degrees, on each side of the middle line, that the period is practically the same, yet when we get to large angles, we know that the period increases, and this varying increase throughout the roll necessarily affects the combination of inclination which comes about when rolling in waves of uniform period. This paper brings to us, I think for the first time, a solution of that difficult problem, and shows it in the diagrams and Plates IX. and X., and gives us really for the first time in a simple form a diagram showing the period of the wave in relation to the angle to which the ship rolls, and the period of swell that causes the maximum oscillation. My Lord, I should like to ask Captain Russo if he could give us at some future time—if, indeed, he has not done it already—a study of the rolling of a ship which has a negative metacentric height to start with. Of course a ship with a negative metacentric height may have positive righting arms at a small angle of heel. That condition of ship is familiar to us: one that has the metacentre in the upright below the centre of gravity, but which, on account of its high side, is perfectly safe at sea. I think the phenomenon of a ship at sea with a negative metacentric height is of some interest, and also of some practical importance. I think wo shall be some time before we can hope to devise a method of modifying the forms of ships so as to enable us to make them better suited for our purpose on the basis of experiment of this kind, because we should need a very great deal of study of the subject not only in rolling itself, but upon the effect on the other elements of design which would come into play the moment we changed the form to meet the required improved conditions of rolling.

Mr. MACFARLANE GRAY (Member of Council): My Lord and Gentlemen, I rise only to speak in compliment of the remarkable machine whose performances, as we witnessed them, have so greatly astonished and pleased us. At our meeting here forty years ago, some of us enjoyed a practical demonstration of the same principles of flotation by the late Dr. Froude—the experiment of the water not overflowing a rocking cup. There was then a large tank with sides partly of glass, and on the water in the tank a wave was set up, and on the surface of the water there floated a tiny raft carrying a small bell in a frame. When on the wave, the clapper still hung in the centre line of the bell, never striking, as if it said to the wave: "Do your utmost, you cannot disturb me, my plumb line is the normal to your surface, however much that surface may be inclined." Eighteen years ago, in a discussion here, I made a remark which is recalled to my mind by this beautiful machine. I explained that the principles of stability of a vessel and the nature of the metacentre were truly illustrated in a child's rocking-cradle. The rocker curve is the curve of buoyancy, and the centre of curvature of the middle part of the curve is the metacentre. Captain Russo has this rocker curve in his machine, and he rocks it upon a plane to which he gives the motion and the inclination of any specified wave. I cannot express to the full the admiration I have for this marvellous mathematical automaton. While looking at it, I said to Lord Glasgow, "What ability in conception, what patience in perfecting, and how excellent the workmanship! And this is the work of a foreigner; these are the men we have to contend with." Let every Englishman, Scot, and Irishman realise this.

Mr. J. I. THORNYCROFT, LL.D., F.R.S. (Vice-President): My Lord and Gentlemen, I should like in a few words to say how much I admire this machine, and that I think it will draw us curves which are most instructive, and likely to lead to very useful results. Captain Russo has solved a most difficult problem. He has reproduced the complete motion of one part of a simple wave, giving the period at the same time, and the correct inclination, with great accuracy. I wish to say how much the Institution is indebted to the author for bringing his machine before us.

Sir NATHANIEL BARNABY, K.C.B. (Vice-President): Might I, my Lord, suggest one thing? Mr. Macfarlane Gray has spoken about contention, but what we have to remark, I think, is the extreme kindness of the Minister of Marine in Italy, in taking care that we should see this apparatus, and I therefore propose that we should ask you to be so good as to convey to him our thanks for his great courtesy.

Mr. A. F. YARROW (Vice-President): I should like, my Lord, to second Sir Nathaniel Barnaby's proposal.

The Right Hon. the Earl of GLASGOW, G.C.M.G. (the President): I am much obliged to Sir Nathaniel Barnaby for proposing it. If no one else had done so, I was prepared to do it myself.

I understand it is the wish of the meeting that I should communicate to the Minister of Marine in Italy our thanks for his kindness in allowing Captain Russo to bring his wonderful machine here for us to see. Now, gentlemen, I think you will all agree with me when I ask you to give Captain Russo a most hearty vote of thanks for his interesting paper, and for allowing us to see his most beautiful apparatus.

Captain G. Russo (Member): My Lord and Gentlemen, it would be a serious task for me to reply to the discussion of my paper, not being as proficient in the English language as I should wish. Nevertheless, the careful manner in which my experimental demonstration has been followed, and the very kind words in which Mr. Froude, Mr. Denny, Professor Biles, Mr. Macfarlane Gray, Mr. Yarrow, and Mr. Thornycroft have expressed their opinions about my poor work, leave me, as my principal duty, the very agreeable one of expressing my most grateful thanks for the welcome given to my paper to-day, which I do very heartily. I am also much honoured by Sir Nathaniel Barnaby's proposition, and I am certain that the Italian Minister of Marine will be very pleased by the vote of thanks addressed to him.

THE STRAINING ACTIONS ON THE DIFFERENT PARTS OF A CRANK SHAFT, ILLUSTRATED BY AN ACTUAL CASE OF A FOUR-CRANKED MARINE SHAFT.

By Professor S. Dunkerley, Associate.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 20, 1902; A. F. Yarrow, Esq., Vice-President, in the Chair.]

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§ 1. The object of this paper is not to bring forward any new theories regarding the strength of crank shafts, but to simply lay on record an actual case of a crank shaft in which the straining actions on the different parts—pins, arms, and shaft—have been worked out from the actual indicator cards. The general methods of determining the various straining actions are familiar to most engineers; but the work and time that have to be spent in order to show how the straining actions at any particular section vary from moment to moment are considerable. The example quoted refers to a modern type of marine crank shaft in which there are four cylinders and four cranks, the centre lines of the cylinders being parallel and in the same plane, and in which the cranks and reciprocating masses are so disposed as to ensure a "balance." The results were primarily intended to illustrate the author's lectures at the Royal Naval College, Greenwich, but it was thought that they might be of sufficient interest to justify bringing them before the members of this Institution. The writer desires to acknowledge his indebtedness to Sir John Durston, K.C.B., R.N., for granting him permission to use, and to Mr. Emdin, Chief Engineer, R.N., for kindly obtaining for him, the necessary data for the paper; and also to Mr. A. H. Roberts, of the Mechanical Department of the Finsbury Technical College, for his valuable assistance in the preparation of the diagrams for reproduction. One of the most interesting and important results of the investigation is that, in the type of engine discussed, the aftermost crank pin is by no means the most severely stressed part of the crank shaft.

Character of Straining Actions.

§ 2. Before considering the case of the four-cranked engine, it might, perhaps, be desirable to briefly refer to the different straining actions on the different parts, so that there will be no difficulty in following the subsequent work.

§ 3. The force acting on any crank pin is simply the effective force transmitted through the connecting rod. The magnitude of this force is slightly greater than the corresponding effective piston pressure on account of the obliquity of the rod. (With a rod four times the length of the crank, the force in the rod only exceeds the force on the piston by about 3 per cent. at most.) It is usually convenient—although, in many cases, by no means necessary, or even desirable—to resolve the force P in the rod into two components—viz.: (1) a force Q perpendicular to the longitudinal plane of the crank arms and shaft, and (2) a force R in the longitudinal plane of the crank arms and shaft (Fig. 1, Plate XI.). The force R induces only a bending moment (and shear) at any section, but the force Q induces a twisting moment in addition. In what follows, in determining the straining actions at any section, we have adopted what is usually termed the "method of sections." Moreover, we assume the force to act on the pin at the centre of its length, instead of being distributed in some manner over the length of the pin. This assumption will over-estimate the bending moment on the pin, and, therefore, errs on the right side.

Case of a Single-Cranked Engine.

§ 4. Consider a single-cranked shaft, supported on two bearings, in which the power is taken off at one end. Let us assume that the bearings exercise no constraint on the shaft.

§ 5. First, consider the forces in the longitudinal plane of the crank arms and shaft. The force R (Fig. 2, Plate XI.) causes reactions R_1 and R_2 at the bearings, the magnitudes of which can be found as in ordinary beams. For convenience, let us consider a shearing force as positive when the left-hand section tends to slide down relatively to the right, so that a bending moment is positive when the forces to the left of the section tend to cause a counter clockwise rotation. Then, in addition to the shear R_1, the part C D of the shaft is subjected to a bending moment which varies uniformly from zero at C to $R_1 \cdot CD$ at D. The arm E D is subjected to a longitudinal force R_1 and a bending moment $R_1 . CD$, which is the same at every section. The pin is subjected to a shear R_1 from E to B, and a bending moment which varies from $R_1 . CD$ at E to $R_1 . CA$ at B, whilst from B to F it is subjected to a shear $R_1 - R$ ($= - R_2$) and a bending at F of $R_1 . CG - R . BF$ ($= + R_2 . GH$). The arm F G is subjected to a bending moment of the same amount (viz., $+ R_2 . GH$) and a longitudinal force $R_1 - R$; whilst the shaft from G to H is subjected to a bending moment which decreases uniformly from $R_2 . GH$ at G to zero at H, together with a shear $R_1 - R$. In fact, the curves of bending moments for the shaft, pin, and arms are as shown in Fig. 3.

§ 6. Next consider the force Q, which acts perpendicularly to the longitudinal plane of the arms and shaft (Fig. 4). Let us consider the twisting moment on the pin and shaft, and also the bending moment on the arms as positive, when, looking from aft in a forward direction, the forces forward of the section tend to cause a clockwise rotation.

(The rotation of the shaft subsequently considered, looking from aft to forward, was in a clockwise direction.) Let the bending moment on the shaft and pin, and also the twisting moment on the arms, be reckoned positive, when, looking along the arms from the pin towards the shaft, the rotation is counter clockwise. Then, for equilibrium, we have reactions Q_1 and Q_2 at the bearings, the magnitudes of which are obtained in the usual way, and a twisting moment T (= Q . A B) applied at the aft end of the shaft. In addition to the shearing force, the part C D is subjected to a bending moment, which varies uniformly from zero at C to Q_1 . C D at D. If we consider any section X of the arm D E, the only force which acts to the left of the section is Q_1, and this causes a couple Q_1 . C X acting in the plane perpendicular to the longitudinal plane of the cranks, whose trace on that plane is C X. The component of this couple in the plane whose trace is E D, viz.: Q_1 . D X, constitutes a bending moment on the arm, whilst the component in the plane whose trace is C D, viz.: Q_1 . C D, constitutes a twisting moment on the arm. Thus the arm D E is subjected to a twisting moment Q_1 . C D, which is the same at every section, and a bending moment which varies uniformly from zero at D to Q_1 . E D at E. If we consider any section of the crank pin, such as Y, the forces to the left of the section are Q_1 and Q, and therefore at Y we have a couple Q_1 . C Y acting in a plane whose trace is C Y, and a couple Q . B Y acting in a plane whose trace is B Y. These are equivalent to a twisting moment Q_1 . B A, and a bending moment Q_1 (A C + B Y) − Q . B Y. Thus the crank pin is subjected to a twisting moment Q_1 . A B, the same at every section, and a bending moment which increases uniformly from Q_1 . C D at E, to Q_1 . C A at A, and then decreases uniformly to Q_1 . C G − Q . F B (= Q_2 . G H) at F. If we consider any section Z of the arm F G, the forces to the left are Q_1 and Q, which give rise to couples Q_1 . C Z and Q . B Z in the planes whose traces are C Z and B Z. Resolving, these give a twisting moment Q_1 . C G − Q . F B (= Q_2 . G H), and a bending moment Q_1 . Z G + Q . Z F. Thus the bending moment varies uniformly from Q_1 . F G at F to Q . F G at G; or, expressed in words, from the twisting moment on the crank pin at the crank pin end, to the twisting moment on the crank shaft at the crank shaft end. This is true, of course, for every crank arm, as is also the statement that the twisting moment on a crank arm is equal to the bending moment on the crank shaft at a section through the arm. The truth of these statements is obvious without the detailed analysis given above. The shaft G H is subjected to a bending moment which varies uniformly from Q_2 . G H at G to zero at H, and to a twisting moment equal at every section to Q . A B. The curves of bending and twisting moments are as sketched (Figs. 4 and 5, Plate XI.), the planes in which the different couples act being sufficiently clear from the diagram.

§ 7. By considering the forces in and perpendicular to the longitudinal plane of the arms, the nature of the straining actions at any section is readily seen. To find the resultant bending moment at any section of the shaft and pin, we may, since these are of circular section, combine the moments in the above two planes. If M_1, M_2 be the bending moments, at any section, in and perpendicular to the longitudinal plane of the cranks, the resultant bending moment at that section is—

$$\sqrt{M_1^2 + M_2^2}$$

At the section B of the crank pin, it is therefore simply—

$$AC \sqrt{R_1^2 + Q_1^2}.$$

This, of course, might have been obtained without resolving the force P into its two components. Thus, in Fig. 6 (Plate XI.), the reactions due to P are the two parallel forces P_1 and P_2. If we consider, say, the section B of the crank pin, the only force to the left of that section is P_1. The couple due to P_1 is equivalent to a bending moment $P_1 . CA$, acting in the plane containing the line of action of P, and parallel to the axis of the shaft (and therefore differing but little from the longitudinal plane of the cylinders), together with a twisting moment acting parallel to the plane containing AB and the line of action of P, and equal in magnitude to P_1 multiplied by the perpendicular distance of A from the line of action of P. The bending moment—

$$P_1 . CA = CA \sqrt{R_1^2 + Q_1^2},$$

since R_1 and Q_1 are the components of P_1, just as R and Q are the components of P. The twisting moment has the same value as $Q_1 . AB$, since the component R in the plane of the crank produces no twisting moment. Thus, in the simple case considered, in order to find the resultant bending and twisting moments at any section of the shaft and pins, it is shorter not to resolve P into its two components. In dealing with the arms we *must* find the moments in, and perpendicular to, the longitudinal plane of the cranks, because the arms are not of circular section, and in more complicated cases we shall find that the resolution is necessary even for the shaft and pins. In any case, resolving the forces into the two components probably makes the character of the straining actions clearer than would otherwise be the case.

§ 8. It is clearly desirable to be able to predict the positions of the crank when the straining actions at any particular section reach their maximum value. The direct bending moment (M) on the pin is a maximum when the force in the connecting rod is a maximum, that is to say (neglecting the slight effect of obliquity), when the effective force on the piston is a maximum. The twisting moment on the pin, in a single-cranked engine, is a constant proportion of the twisting moment due to the cylinder, viz., the proportion—

$$\frac{Q_1}{Q} = \frac{AH}{CH}$$

(since the twisting moment on the pin is $Q_1 . AB$ and of the cylinder $Q . AB$); it is therefore a maximum when the twisting moment T of the cylinder is a maximum, which, of course, need not by any means be when the piston pressure is a maximum. The *direct* stress induced in the pin has the same *magnitude* as the shear stress calculated from what is usually termed the equivalent twisting moment, and, therefore, to obtain the maximum value of the direct stress, we require the position of the crank which gives us the maximum value of—

$$M + \sqrt{M^2 + Q_1^2 . \overline{AB}^2},$$

or of—

$$M + \sqrt{M^2 + T^2 . \left(\frac{AH}{CH}\right)^2}.$$

This position of the crank cannot definitely be predicted. When the crank is in the position corresponding to maximum piston pressure, M (very approximately) has its maximum value, but T has not, perhaps, reached its maximum. As the crank rotates M will decrease and T, probably, increase; so that whether the above expression increases or decreases can only be found by calculating its value for a number of positions of the crank. Probably it will be found—more particularly since M appears twice in the above expression—that the increase in T is not sufficient to counterbalance the decrease in M, so that as a first approximation we may say that the maximum equivalent twist will take place in the position of the crank for which the force on the piston is a maximum (see § 25).

§ 9 So far as the arms are concerned, the twisting moments on them depend on the values of the reactions perpendicular to the longitudinal plane, and have, therefore, their maximum value when these reactions are a maximum; that is to say, when Q is a maximum, or when the crank is in that position for which the twisting moment is a maximum. This, likewise, is the position of the crank when the bending moments on the arms, in the plane perpendicular to the longitudinal plane of the cranks, have their maximum value. The bending moments on the arms in the longitudinal plane are a maximum when the component of the force in the rod resolved along the crank arm has its maximum value.

Case of a Multiple-cranked Engine when the Effect of Continuity of Shaft is neglected.

§ 10. In the above we have assumed that we have a single-cranked engine in which the power is absorbed at one end. Now, suppose that the crank considered is one of the cranks of a multiple-cranked engine and that the forward cylinders transmit a combined twisting moment T_1. Let us further assume that each part of the crank shaft—from journal to journal—may be considered an independent beam, or, in other words, let us assume that the bending moment at each journal is zero. Then the straining actions on the different parts due to the piston pressure of the cylinder considered can be determined in exactly the same manner as before, and the effect of the twisting moment T_1 from the previous cylinders, is to simply increase the twisting moment on the pin and shaft, as well as the bending moment on the arms in the plane perpendicular to the longitudinal plane of the cranks, by the amount T_1 (Fig. 6, Plate XI.). The total bending moment on the shaft and pin, and also the bending moment on the arms in the longitudinal plane of the cranks, as well as the twisting moments on the arms, will be unaffected by T_1, provided, of course, that each part of the shaft may be considered a separate beam. Thus the direct bending moment, M, on the pin is a maximum when the force on the piston is a maximum; the direct shear stress on the pin is a maximum when—

$$T_1 + T \cdot \frac{AH}{CH}$$

is a maximum; the resultant direct stress on the pin is a maximum when—

$$M + \sqrt{M^2 + \left(T_1 + T \cdot \frac{AH}{CH}\right)^2}$$

is a maximum; the bending moment, perpendicular to the longitudinal plane of the cranks, is a maximum on the forward arm when—

$$T_1 + T \cdot \frac{AH}{CH}$$

is a maximum, and on the aft arm, when $T_1 + T$ is a maximum; the bending moment on the arms in the longitudinal plane of the cranks is a maximum when the component of the force in the rod resolved along the crank radius is a maximum; and the twisting moment on the arms is a maximum when T is a maximum.

Case of Multiple-cranked Engine when the Effect of the Continuity of the Shaft is considered.

§ 11. Now, as a matter of fact, the assumption just made, viz., that the shaft is subjected to no bending moment at any of the journals, although it simplifies the problem considerably, cannot be strictly true. The shaft is in the condition of a continuous beam carrying concentrated loads, and the reactions at the main bearings might be quite different from those previously assumed. The precise determination of those reactions is a matter of great difficulty, but, assuming them to have been determined, the methods already given may be applied—always remembering that in taking any section we must consider *all* the forces to the left of the section. Thus, for simplicity, take the case of a multiple-cranked engine in which the cranks are parallel, and resolve the forces along each of the rods (the rods need not, of course, be parallel) into two components, Q and R, perpendicular to the longitudinal plane of the cranks, and in the longitudinal plane, as before. Consider, first, the forces in the longitudinal plane (Fig. 7, Plate XI.). The forces R_1, R_2, R_3 . . . will give rise to reactions R_a, R_b . . . at the different journals, the magnitudes of which will depend upon the forces R_1, R_2 . . . (The directions of all the reactions have been drawn opposite to the forces, but this need not be the case, nor need the forces all act in the same direction.) The shaft, arms, and pins are, in this plane, subjected only to a bending moment (neglecting the shearing force). If we assume that the bending moment at the outermost journal A is zero, the bending moment at any section, such as X, is clearly

$$
\begin{aligned}
& R_a . AX - R_1 . GX + R_b . BX - R_2 . HX + \ldots + R_d . DX \\
&= R_a . AD - R_1 . GD + R_b . BD - R_2 . HD + \ldots \\
&\qquad + (R_a - R_1 + R_b - R_2 + \ldots + R_d) . DX \\
&= M_d + F_d . DX, \qquad (1)
\end{aligned}
$$

in which M_d is the bending moment at the journal D, and F_d the shearing force just to the right of D. Knowing, therefore, the bending moments and the shearing forces at the different journals, we can at once get the bending moment at any section of the shaft, pin, or arms. If we assume the crank shaft to behave like a beam of uniform section, in which the points of support are all on the same level, the Equation of Three Moments for concentrated loads can be readily shown to be given by the formula*—

$$
\begin{aligned}
& M_a l_1 + M_c l_2 + 2 M_b (l_1 + l_2) \\
&= R_1 l_1^2 k_1 (1 - k_1)(1 + k_1) + R_2 l_2^2 k_2 (1 - k_2)(2 - k_2), \qquad (2)
\end{aligned}
$$

in which the different symbols have the meanings ascribed to them in Fig. 8, and in which M_a, M_b, M_c are the bending moments at the points of supports A, B, and C. This equation applies to any two adjacent spans, so that if we have (as in Fig. 7) five spans, we get four equations between the six unknown quantities M_a, M_b, M_c, M_d, M_e, and M_f. To find their values we must, therefore, make two assumptions. We might assume that the values of M_a and M_f are zero, or that the shaft is constrained in direction at the ends A or F. The first assumption is the simpler and, on the whole, possibly the more accurate. It is the assumption made in dealing with

* It must not be taken for granted that this formula strictly applies to such a structure as a crank shaft. The bending of the arms themselves, and the method of their attachment to the shaft and pins, might materially affect the problem. In addition, the pins and shaft need not be of the same size. The equation given, therefore, must only be looked upon as giving a very good approximation to the effect of continuity of the shaft. A rougher approximation will be given in § 14, but these objections are equally applicable to this rough approximation as to the above.

the actual crank shaft considered later on. Having thus determined the bending moments, the shearing force to the right of any bearing, such as D, is then given by taking moments about the next bearing, such as E, giving us an equation like—

$$M_e = M_d + F_d \,.\, D\,E - R_4 \,.\, K\,E, \qquad (3)$$

from which F_d may be determined. In this manner the shearing force to the right of any of the journals can be found, and so also the bending moment, M_x, at any section X of the shaft or pin can be determined by equation (1).

§ 12. Now, consider the forces, Q, perpendicular to the longitudinal plane of the cranks (Fig. 9, Plate XI.). If we consider any section X of the shaft, the reactions Q_a, Q_b . . . cause only a bending moment, whilst Q_1, Q_2 cause twisting as well as bending. Thus the bending moment at X, due to Q_2 is $-Q_2 \,.\, H\,X$, whilst the twisting moment is $Q_2 \times$ crank radius. The bending moment at X is therefore obtained exactly as before, with the exception that for R we substitute Q, and this applies equally for the pin as for the shaft. The twisting moment at X is, of course, merely the combined twist due to the cylinders forward of the section considered. The twisting moment on any arm such as N O is equal to the bending moment on the shaft at the section N. The bending moment at N of the arm N O, in the plane perpendicular to the longitudinal plane of the cranks, is (considering the forces to the left of N) equal to $(Q_1 + Q_2 \ldots) \times$ crank radius, that is, is equal to the twisting moment on the shaft at N; whilst the bending moment at the end O is

$$\begin{aligned} &(Q_a + Q_b + \ldots) \,.\, N\,O \\ &= (Q_1 + Q_2 + \ldots)\, N\,O + (Q_a + Q_b + \ldots - Q_1 - Q_2 - \ldots)\, N\,O \\ &= \text{twisting moment from the previous cylinders} \\ &+ \text{shearing force at section N} \times \text{crank radius.} \qquad (4) \end{aligned}$$

This last expression is, of course, also the twisting moment on the crank pin. The bending moment on the aft crank arm varies from the twisting moment on the crank pin, at the crank pin end, to the total twisting moment due to all the previous cylinders (including the one under discussion) at the shaft end.

§ 13. Now, in the general case, the cranks are not parallel. In any position of the crank shaft, such as is represented by the end view in Fig. 10, if we still consider the shaft to act as a continuous beam, we must resolve the force P, in each rod into two components, V and W, acting parallel to any two perpendicular planes; and we must consider the effects of each of these two sets of forces separately. Probably the most convenient planes are the longitudinal plane of the cylinders and the plane through the crank shaft perpendicular to this longitudinal plane. The isometric view (Fig. 11, Plate XI.) shows the components taken parallel to these planes, and the longitudinal plane has been taken vertical in order to illustrate the shaft subsequently considered. The components of the forces at the crank pins parallel to the longitudinal plane are simply the effective pressures on the different pistons, whilst the components perpendicular to this plane are equal to the guide reactions. The forces in the longitudinal plane are, therefore, very much the more important. Both sets of forces will cause bending and twisting moments on the shaft, arms, and pins. Considering one of the sets of forces, say the V-forces, since these are all parallel, the reactions at the different journals are likewise parallel. Their magnitudes, and also the magnitudes of the bending moments on the journals, are determined in exactly the same way as described before for the Q and R forces. Thus, at every section of the shaft and pins, we can calculate the bending moment and shearing force in the two projection planes chosen, so that the resultant bending moment at any section is given by the square root of the sum of the squares of the two components. The twisting moment at any section of the shaft is, of course, simply the combined twisting moment of the cylinders forward of the section considered. The twisting moment at any section of a pin is the algebraical sum of the twisting moment due to the cylinders forward of the pin considered, and of the moment of the shearing forces, in the two planes of projection at the section considered, taken about the crank pin. This follows immediately from equation (4). The bending moment on a crank arm, perpendicular to the plane of the arm and shaft, varies uniformly from the twisting moment on the shaft at the shaft end of the arm to the twisting moment on the pin at the pin end of the arm. The bending moment on an arm, *in* the plane of the arm and shaft, as well as the twisting moment on the arm, is the same at every section of the arm, and is determined as follows:—At the shaft end of any arm we can find the bending moments in the two projection planes, that is to say, in the two perpendicular planes which intersect along the axis of the shaft. The algebraic sum of the components of these couples, in the plane of the arm and shaft, constitutes the bending moment on the arm, and in the plane perpendicular to this plane and which contains the axis of the shaft, constitutes the twisting moment. Attention must, of course, be paid to sign in resolving the couples. These statements will be clear from an examination of Fig. 9 combined with the preceding analysis. A more detailed analysis is given later. Following the convention as regards sign given in §§ 5 and 6, in Fig. 11 (Plate XI.), down forces in the longitudinal plane, and forces acting from right to left (looking from aft) in the second plane of projection, must be considered positive.

Rough Approximation to Determine the Constraining Effect of the Bearings.

§ 14. The first assumption made in determining the straining actions, viz., assuming that there is no bending moment on the journals and that the twisting moment alone is transmitted from cylinder to cylinder, enables the different straining actions to be readily calculated, but will, in general, overestimate the straining actions on the pins and underestimate those on the journals—a very close approximation to the true straining actions being given by the method just described. A rough approximation to determine the constraining effect of the bearings is to assume that each bearing, instead of merely supporting the shaft, effectually constrains it in direction, so that each part of the shaft is in the condition of a beam, carrying a concentrated load, and fixed in direction at each end, each section acting as an independent beam whilst transmitting the twisting moment. This assumption would involve a bending moment on each journal, but would make the bending moment on the pin very much less than that estimated on the first assumption. It would, in fact, generally underestimate the straining actions on the pins and overestimate those on the journals. Thus, for example, if the load P were applied midway between the bearings, on the first assumption the bending moment on the pin would be $\frac{P\,l}{4}$ and, on each journal, zero; on the second assumption, the bending moment on the pin and the two journals would be the same and equal to $\frac{P\,l}{8}$, assuming l to be the same in each case (see § 34). The reactions in both cases would be the same, so that the twisting moments on the pins and shaft would be unaltered. Thus, if we estimate the straining actions on the first assumption, we can readily modify them to fit in with the third assumption, and so compare the results with the correct results, when the effect of continuity is considered in the manner described above.* The alterations in the straining actions on the arms could be found by the methods given above.

The Analysis of the Actual Crank Shaft.

§ 15. The crank shaft chosen, as already pointed out, is that of a four-cranked marine engine of the inverted type, in which the reciprocating masses and cranks are so disposed as to ensure a balance. The longitudinal view of the crank shaft is shown in Fig. 12 (Plate XII.), and the end view, looking from aft in a forward direction, in Fig. 13. The crank angles, direction of rotation, and centre line of the shaft, &c., with the leading dimensions are shown in Figs. 14 and 15. The stroke of all the cylinders is 4 ft., and the length of all the connecting rods 8 ft. The remaining leading data are given in Table I.

§ 16. The case considered corresponds to a full-power trial when the engines were running at 124·2 revolutions per minute, and the combined horse-power of the four cylinders 10,300. The only straining forces on the crank shaft are taken to be those due to effective piston pressure, the forces necessary to drive the valves, &c., being neglected.

* See remarks in the footnote to § 11.

TABLE I.

PRINCIPAL DATA.

Stroke, 4 ft. Length of Connecting Rod, 8 ft. Speed, 124·2 revolutions per minute.

Cylinder.	H.P.	M.P.	F.L.P.	A.L.P.
Diameter of piston in inches	36	59	68	68
Diameter of piston rod in inches	9·5	9·5	7·5	7·5
Effective area at top end of cylinder in square inches	1,018	2,734	3,632	3,632
Effective area at bottom end of cylinder in square inches ...	947	2,663	3,587	3,587
Total weight (in lbs.) of purely reciprocating parts (*i.e.*, piston, piston rod, and crosshead complete)	5,740	6,470	5,980	5,980
Weight (in lbs.) of connecting rod complete	8,190	8,190	4,600	4,600

The indicator cards are shown in Figs. 16 to 19 (Plate XII.), the full line referring to the bottom end, and the dotted line to the top end of the cylinder. The mean effective steam pressures and horse-powers are given in Table II.

TABLE II.

POWER DEVELOPED IN THE DIFFERENT CYLINDERS.

Cylinder.	H.P.	M.P.	F.L.P.	A.L.P.
Mean effective pressure at bottom end in lbs. per square inch	110·6	44·8	16·42	17·07
Mean effective pressure at top end in lbs. per square inch	108·2	40·2	15·92	17·02
Mean effective pressure for the two ends in lbs. per square inch	109·4	42·5	16·17	17·04
Horse-power developed	3,240	3,450	1,760	1,850

Very approximately, therefore, the power developed in each of the H.P. and M.P. cylinders is one-third, and in each of the L.P. cylinders one-sixth of the total.

EFFECTIVE PISTON FORCES.

§ 17. In order to get the effective force transmitted to the crank pin for any position of the piston, we must first take the difference of the forces due to steam pressure on the two sides of the piston. Thus, taking the H.P. cylinder, when the piston is moving up, and has moved through one-quarter of the stroke from the bottom end, the gauge pressure in pounds per square inch is 232 on the bottom side, and 85 on the top side of the piston. The effective up force due to steam pressure is, therefore, 232 × 947 − 85 × 1,018, that is 133,000 lbs., or 59·5 tons. In this way we obtain Figs. 20–23 (Plate XII.). In all these figures the ordinates to the curves U U and D D, reckoned from the "steam" line S S, give the total effective up or down force due to steam pressure, in tons, on the piston in any position. Ordinates above S S correspond to an up force, and below S S to a down force.

Modification due to Weight of Moving Parts.

§ 18. The second thing that we have to do is to modify these forces on account of the weights of the moving parts. The purely reciprocating parts clearly cause a down force in all positions of the piston equal to their weight. The effect of the weight of the connecting rod, in modifying the force transmitted to the crank pin, is not so simple; but very approximately, so far as the *weight* of the rod is concerned, we may take the effect to be the same as if a weight equal to that of the rod were concentrated at the crosshead pin.* In that case, in the high pressure cylinder, for example, the total down force due to the weights of the moving parts, will be 5,740 + 8,190 = 13,930 lbs. = 6·22 tons. We need, therefore, only to shift our datum line up through this distance, and so get the "weight" line W W as our new datum from which to measure our up or down forces. In the M.P. cylinder the vertical shift of the datum line is 6·54 tons, and in the two L.P. cylinders 4·71 tons.

Modification Due to Inertia of Moving Parts.

§ 19. The third thing is to correct these forces on account of the inertia of the moving parts. The correction necessary for the purely reciprocating parts is a simple matter. If, in any position, the acceleration of the piston is f feet per second per second, the force necessary to cause this acceleration is f multiplied by the mass (*i.e.*, weight ÷ 32·2) of the reciprocating parts. The

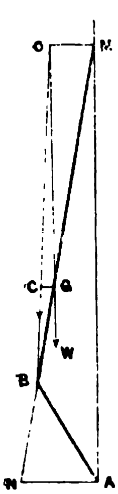

FIG. A

* This, of course, refers only to an engine in which the pistons move vertically. To determine the precise effect on the turning moment on the crank shaft, the weight W (Fig. A) of the rod causes reactions at the crosshead and crank pins; neglecting friction, that at the crosshead must be perpendicular to the guides, and, therefore, that at the crank pin must act along O B. The force along O B can be found by the triangle of forces, *i.e.*, by drawing a triangle whose sides are parallel to O G, O M, O B. In fact, if G C be drawn perpendicular to the stroke, then, if O G represent W, O C is the force on the crank pin. The moment of that force about A is the turning moment on the shaft due to the weight of the rod. If the weight W were concentrated at the crosshead pin, the moment would have been W . A N, which differs but slightly from the true moment, the point N being the point in which the connecting rod produced meets the perpendicular through A to the line of stroke.

motion of the connecting rod is, however, complicated, but it may be shown* that, so far as the effect on the crank effort diagram is concerned, we may very approximately reproduce it by imagining a certain fraction of its mass to move with the purely reciprocating parts. That fraction is the ratio of the distance of the centre of gravity of the rod from the crank pin to the total length of the rod (*i.e.*, $\frac{BG}{BM}$ in Fig. A), and, in the case considered, is ·38. Thus, in the four cylinders, the weight of the reciprocating parts + ·38 of the weight of the rod will be 8,850, 9,580, 7,730, and 7,730 respectively, and, therefore, the mass (*i.e.*, weight ÷ 32·2) will be 275, 298, 240, and 240 respectively. These numbers multiplied by the acceleration of the piston will give us the inertia force which has to be subtracted from or added to our previous results according as f is positive or negative (that is to say, according as the piston is being accelerated or retarded). At the bottom and top ends of the stroke, f has the values (assuming the crank shaft to rotate uniformly)—

$$\omega^2 r \left(1 \mp \frac{1}{n}\right),$$

in which ω is the angular velocity of the crank shaft in radians per second, r the crank radius in feet, and n the ratio of the length of the connecting rod to the crank radius. In any other position it is very approximately given by the formula—

$$\omega^2 r \left(\cos\theta - \frac{\cos 2\theta}{n}\right),$$

in which θ is the angle turned through by the crank measured from the bottom dead centre.† In the case under discussion, $n = 4$, and the revolutions per minute 124·2, so that $\omega = 13$, $\omega^2 = 169$, and $\omega^2 r = 338$. Thus on the up-stroke the acceleration is + 253 ft. per second per second at the bottom end, and − 422 at the top end. In the down-stroke it is + 422 at the beginning and − 253 at the end of the stroke. The force necessary to cause the acceleration in the up-stroke, or the retardation in the down-stroke, is, therefore, in the high-pressure cylinder, 275 × 253 = 69,575 lbs. = 31 tons at the bottom end; whilst the corresponding quantity at the top end is 51·6 tons. When the piston has moved through one quarter of the up-stroke, measured from the bottom end, $\theta = 66°$, and the acceleration is, therefore, 193 ft. per second per second, and the corresponding inertia force, in the high-pressure cylinder, is 53,000 lbs., or 23·7 tons. In the early part of each stroke, the effective force transmitted to the crank pin is reduced, and in the latter part increased. The correction for inertia is, therefore, made by erecting on the line W W a line the ordinates to which represent the force of acceleration in the different positions of the piston. It is repre-

* See *Engineering*, June 2, 1899 (Vol. LXVII., p. 695), "On the Connecting Rod Problem," by Prof. Dunkerley.

† For a convenient geometrical construction involving this formula, see the above article.

sented by the line I I in Figs. 20 to 23 (Plate XII.). Thus for the H.P. cylinder, the ordinate at the bottom end, reckoned from W W, is 31 tons, and at the top end is − 51·6 tons. The ordinates between the curves U U, D D, and the "inertia" line I I may now be taken to represent the effective up or down force transmitted from the piston to the crank pin: an up-force if the curves lie above I I, a down-force if they lie below it. It will be noticed that in the L.P. cylinders—particularly at the commencement of the down-stroke—the force necessary to cause the acceleration of the moving parts is greater than that available from the steam and weight forces, with the result that near the beginning of the stroke the crank shaft drives the piston.

Twisting Moments on the Different Parts of the Shaft.

§ 20. The next step in the analysis is to determine the twisting moments on the different parts of the crank shaft as the crank rotates. The results are shown in Figs. 24, 25, 26 (Plate XIII.). The turning moment due to the effective force on any piston is obtained by multiplying that effective force by the distance A N in Fig. A (§ 18). Thus, for example, in the H.P. cylinder, when the piston has moved through one-quarter of the up-stroke, the effective piston pressure, from Fig. 20, is 29·6 tons, and the intercept A N is 1·67 feet. The turning moment on the crank shaft is therefore 49·4 foot tons, the crank angle which the H.P. crank makes with the bottom dead centre being 66°. (In this position the F.L.P. crank makes an angle of 261° with its bottom dead centre.) The various twisting moment curves have been plotted on a crank angle base, the angle in all cases being determined by the angle which the F.L.P. crank has turned through from its bottom dead centre. The turning moment diagrams due to the separate cylinders have been placed at the proper angle difference with respect to the F.L.P. crank; the angle differences being given by Fig. 14 (Plate XII.). The turning moment curves due to the F.L.P. and H.P. cranks separately are shown by the faint curves in Fig. 24, that due to the M.P. cylinder alone by one of the faint curves in Fig. 25, and that of the A L.P. cylinder alone by one of the faint curves in Fig. 26 (Plate XIII.). Thus, for the H.P. curve, the ordinate at an angle 261° is 49·4 foot tons. Proceeding from forward to aft, the thick line in Fig. 24 represents the combined twisting moment due to the F.L.P. and H.P. cylinders. That curve is reproduced (faintly) in Fig. 25, and the thick line of that figure represents the twisting moment due to the F.L.P., H.P., and M.P. cylinders. This last curve is reproduced in Fig. 26, and, combined with the twisting moment of the A.L.P. cylinder, gives us the thick line of Fig. 26, which represents the turning or twisting moment transmitted to the propeller shaft. The curves give us the twisting moment at any particular journal for all positions of the cranks. The principal results are given in Table III. It will be noticed that in the propeller shaft the maximum twisting moment is 1·16 times the mean and the minimum ·8 times the mean.

TABLE III.

TWISTING MOMENTS ON SHAFT, IN FOOT TONS.

Cylinders.	F.L.P.	H.P.	M.P.	A.L.P.	F.L.P + H.P.	F.L.P.+H.P. +M.P.	F.L.P.+H.P.+ M.P.+A.L.P.
Maximum	87·6	138·0	138·0	88·4	202·0	230·0	226·0
Mean	33·2	61·0	65·0	35·0	94·2	159·0	194·2
Minimum	−9·0	0·0	0·0	−10·0	−7·5	98·0	155·0
Ratio of maximum to mean	2·64	2·26	2·13	2·52	2·14	1·45	1·16
Ratio of minimum to mean	−0·27	0·0	0·0	−0·29	−0·08	·62	·80
Ratio of maximum to the mean twisting moment on the propeller shaft	·45	·71	·71	·45	1·04	1·18	1·16

BENDING MOMENTS ON THE PINS.

§ 21. The next thing to consider is the bending moment on the various crank pins. The assumption that we first make is that the bending moment on the shaft at each of the journals is zero, so that only the twisting moment is transmitted from crank to crank. It at once follows that if P be the force transmitted along the rod in any position of one of the cranks, then, referring to Fig. 6 (Plate XI.), the bending moment on the pin is greatest at the section where the load, supposed to be concentrated, comes on the pin. (Calculating the bending moment on the assumption of a concentrated load gives us a slightly greater value than if we assumed it distributed.) Its value would be given by the expression $P_1 \cdot AC$ or $P \cdot \frac{AC \cdot AH}{CH}$, which, for any cylinder, would be a constant multiple of P. The value of P, for any position of the piston, may be taken to be slightly greater, on account of obliquity of the rod, than the corresponding effective pressure on the piston,* and can therefore be readily estimated from Figs. 20 to 23 (Plate XII.). Thus, for example, in the H.P. cylinder (measuring distances from the centres of the journals) we have (from Fig. 15) $AC = 3{\cdot}33$ ft., $AH = 3{\cdot}22$ ft., and $CH = 6{\cdot}55$ ft., so that the bending moment on the crank pin is $1{\cdot}638$ P foot tons, P being in tons. When the H.P. piston has moved through one-quarter of the up-stroke, the effective

* Here we are assuming that the whole force on the pin acts in the direction of the rod. This is not strictly true so far as the forces due to the weight and inertia of the rod are concerned, but the difference is very small. Thus in Fig. A (§ 18), the force on the pin due to the weight of the rod acts along O B instead of M B.

force on the piston is 29·6 tons, so that the force in the rod would be about 30·5 tons, and, therefore, the bending moment on the pin, in this position, $1{\cdot}638 \times 30{\cdot}5 = 50{\cdot}0$ foot tons. The angle through which the H.P. crank has turned from the bottom dead centre in the direction of rotation is 66°, so that, by proceeding in this way, we can plot a curve of bending moments for the pin on a crank angle base. This has been done for the four cylinders, and the results are shown in Figs. 27 to 30 (Plate XIII.). In interpreting these diagrams, it should be remembered that the plane of the bending moment is that which contains the centre line of the connecting rod and is parallel to the longitudinal plane of the shaft, and therefore differs but little from the longitudinal plane of the cylinders. Barring obliquity, the shape of the curves would be exactly similar to curves of effective piston pressure, plotted on a crank angle base. Following the convention as regards sign in § 13, when the force in the rod is an up-force (*i.e.*, the rod is in tension), the bending moment has been considered positive; when a down-force, negative. In all cases, the angles given are the angles which the crank considered has turned through from its bottom dead centre.

Twisting Moments on Crank Pins.

§ 22. As regards the twisting moment on the crank pin, its magnitude depends upon the twisting moment transmitted from the forward cylinders, and also upon the reaction at the forward journal of the cylinder considered. Referring to Fig. 4 (Plate XI.), the twisting moment due to the reaction is equal to $Q_1 \times$ crank radius, *i.e.*, to $Q \times$ crank radius $\times \frac{\mathrm{A\,H}}{\mathrm{C\,H}}$ = twisting moment due to the cylinder considered $\times \frac{\text{-}}{\mathrm{C\,H}}$. For any cylinder this multiplier is constant, so that the twisting moment on the crank pin due to the cylinder considered is represented by a curve exactly similar in shape to the twisting moment curve of that cylinder. For example, in the high-pressure cylinder—

$$\mathrm{A\,H} = 3{\cdot}22, \quad \mathrm{C\,H} = 6{\cdot}55, \quad \text{and} \therefore \frac{\mathrm{A\,H}}{\mathrm{C\,H}} = {\cdot}492.$$

When the H.P. crank has turned through an angle of 66° from its bottom dead centre (corresponding to one-quarter of the up-stroke of the piston), the turning moment due to the high-pressure cylinder is 49·4 foot tons, and therefore the twisting moment on the H.P. pin, due to the reaction, is $\cdot 492 \times 49{\cdot}4 = 24{\cdot}2$ foot tons. In this position of the H.P. crank, the F.L.P. crank has turned through an angle of 261° from its bottom dead centre, and therefore, from Fig. 24 (Plate XIII.), the twisting moment transmitted from the F.L.P. crank is 70·0, giving a total twisting moment on the H.P. pin of $24{\cdot}2 + 70{\cdot}0 = 94{\cdot}2$ foot tons. By proceeding in this way, we can get the total twisting moment for different positions of the crank, and plot the results on a crank angle base. The curves for the four cylinders are shown in Figs. 31 to 34 (Plate XIII.). Fig. 31 refers to the F.L.P. crank. Since no twisting moment is transmitted forward of this crank, the total twisting moment on the pin is that due to the reaction on the forward journal. Fig. 32 refers to the H.P. cylinder. The ordinates to the dotted line give the twisting moment on the pin due to the reaction on the forward journal; the chain-dot line is the curve of twisting moments from the forward cylinders, *i.e.*, in this case from the F.L.P. cylinder, the curve being set at the proper phase angle difference with respect to the H.P. crank; whilst the full line represents the total twisting moment on the pin, obtained by taking the algebraic sum of the ordinates of the two preceding curves. Thus at 66°, the ordinates to these three curves are respectively 24·2, 70·0, and 94·2 foot tons. Figs. 33 and 34 are similar to Fig. 32, the dotted line being the twisting moment on the pin, due to the reaction on the forward journal of the cylinder considered, the chain-dot curve representing the twisting moment transmitted from the previous cylinders (in Fig. 33, it is the twisting moment curve of the F.L.P. + H.P. cylinders from Fig. 24; in Fig. 34, the twisting moment curve of the F.L.P. + H.P. + M.P. cylinders from Fig. 25), and the full line giving the total twisting moment on the pin. The twisting moment is considered positive when it acts in the same direction as the direction of rotation of the engine, and in all cases, the angle is the angle turned through by the crank, considered from the bottom dead centre.

Equivalent Twisting Moments on Crank Pins.

§ 23. Having thus obtained the bending moment (M) and twisting moment (T) on the crank pin, we can now find the equivalent twisting moments which may be used in order to calculate the maximum shear and the maximum direct stress induced in the crank pin. The former is given by

$$\sqrt{M^2 + T^2},$$

and the latter by

$$M + \sqrt{M^2 + T^2},$$

the latter being the more important of the two. The magnitude of the shear stress derived from a twisting moment of magnitude equal to

$$M + \sqrt{M^2 + T^2},$$

is equal to the *magnitude* of the maximum *direct* stress in the pin. We may represent these moments by curves which are shown in Figs. 35 to 38 (Plate XIII.), and which are derived from the previous figures by taking corresponding values of M and T. Thus, in the H.P. crank, when the angle is 66°, M = 50, T = 94·2, and therefore

$$\sqrt{M^2 + T^2} = 107 \text{ foot tons nearly,}$$

and

$$M + \sqrt{M^2 + T^2} = 157 \text{ foot tons.}$$

The dotted curves (in Figs. 35—38) give the *magnitudes* of

$$\sqrt{M^2 + T^2},$$

and the full curves the *magnitudes* of

$$M + \sqrt{M^2 + T^2}$$

for a complete revolution of the different cranks.

§ 24. These curves well repay study. Since the four pins have all the same dimensions, the ordinates to these curves are proportional to the intensities of stress induced. As we proceed aft, the more uniform do the curves become, that is to say, the less the variation of the stress induced as the crank shaft rotates. But the maximum direct stress increases by no means progressively as we proceed aft. In point of fact, the H.P. pin is the most severely stressed pin, then the A.L.P. pin, and, following closely, the M.P. pin. In noticing these results, we must not forget that the four cylinders were not designed for equal power, but that each of the two outer cylinders develops only about half the horse-power of either of the other two. Since, therefore, the distances between the journals of the different cylinders are not very different, the bending moments on the H.P. and M.P. pins are greater than on the L.P. pins, so that although the twisting moment on the H.P. pin is less than on the A.L.P. pin, yet the equivalent twisting moment on the former is greater than on the latter. In any case, this result can only be taken to apply to the type of engine considered, in which the two outer cylinders develop much less power than the two inner ones.

§ 25. The principal results are given in Table IV. The maximum values, with the corresponding crank positions, of

$$M, \quad T, \quad \sqrt{M^2 + T^2}, \quad \text{and} \quad M + \sqrt{M^2 + T^2}$$

are given in Nos. 1 to 8. The first, as already pointed out, takes place when the piston pressure is a maximum, but the maximum value of T by no means need be simultaneous with the maximum value of M. The position for which T is a maximum not only depends on the cylinder considered, but upon the combined twisting moments of the previous cylinders. It will be noticed that the positions of the crank for which

$$M + \sqrt{M^2 + T^2}$$

is a maximum are practically, in all the four cylinders, the same as when M is a maximum. In order to make further comparison, that position of the crank has

TABLE IV.

MOMENTS ON PINS, IN FOOT TONS.

No.	Cylinders.	F.L.P.	H.P.	M.P.	A.L.P.
1	Maximum direct bending moment on pin	79	129	132	80
2	Angle that crank has turned through, from bottom dead centre	300° or 150°	290°	292°	165° or 307°
3	Maximum direct twisting moment on pin	45	136	215	225
4	Angle from bottom dead centre	285° or 132°	290°	30°	195°
5	Maximum value of $\sqrt{M^2+T^2}$ on pin ...	89	187	222	226
6	Angle from bottom dead centre	290°	290°	30°	180°
7	Maximum value of $M+\sqrt{M^2+T^2}$ on pin	167	316	283	300
8	Angle from bottom dead centre	300°	290°	240° or 290°	173°
9	Ratio of the maximum equivalent twisting moment to mean twisting moment on the propeller shaft	·86	1·63	1·46	1·54
10	Angle when piston pressure is a maximum	300°	290°	292°	165°
11	Maximum piston pressure, in tons ...	51·9	76·0	78·0	52·3
12	Bending moment on pin	79	129	134	80
13	Twisting moment on pin...	39	136	73	200
14	Value of $\sqrt{M^2+T^2}$ on pin	88	187	150	215
15	Value of $M+\sqrt{M^2+T^2}$ on pin	167	316	283	294

been taken for which the effective pressure on the piston is a maximum, and the values of

$$M,\ T,\quad \sqrt{M^2+T^2},\ \text{and}\quad M+\sqrt{M^2+T^2}$$

corresponding to this position have been scaled off. If we compare these results with the maximum values of these expressions (that is to say, compare No. 10 with No. 8, and No. 15 with No. 7), we shall find the agreement remarkably close, thus justifying the statement in § 8, that the equivalent twisting moment may be taken to be a maximum in that position for which the effective piston pressure is a maximum. The values of the maximum equivalent twisting moment on the different pins (Nos. 7 and 9) as we proceed aft, ought to be noticed.

Moments on the Arms.

(1) *Twisting Moments.*

§ 26. Having considered the shaft and the pins, the next thing is to consider the arms. For any cylinder, the after arm is the one which is usually the more severely strained, so that this is the arm to which the following results apply. The arm is subjected to both twisting and bending, and, since it is not of circular section, we must consider the bending moment in the longitudinal plane of the arms and shaft, and in the plane perpendicular to this longitudinal plane, separately. The method of determining the different straining actions has been already described. The twisting moment on the after arm—still assuming that the bending moments at the different journals are zero—is (Fig. 4, Plate XI.) $Q_1 . CG - Q . FB$, which is equal to $Q_2 . GH$, and, therefore, to

$$Q . \frac{AC . GH}{CH},$$

in which Q is the component of the force in the rod, resolved perpendicularly to the crank radius, and is consequently proportional to the twisting moment of the cylinder considered. Thus, in the H.P. cylinder (see Fig. 15, Plate XII.)—

$$AC = 3{\cdot}33 \text{ ft.} \quad GH = 1{\cdot}70 \text{ ft.}, \quad CH = 6{\cdot}55 \text{ ft.},$$

and, therefore, the twisting moment on the after arm is $Q \times {\cdot}865$. Since the crank radius is 2 feet, the twisting moment is equal 2 Q, and, therefore, the twisting moment on the after arm is equal to the twisting moment due to the H.P. cylinder $\times {\cdot}432$. Thus, for example, when the crank has turned through 66° from the bottom dead centre (corresponding to one-quarter of the up-stroke), the turning moment of the H.P. cylinder is 49·4 foot tons, and, therefore, the twisting moment on the after arm is 21·3 foot tons. Thus, by proceeding in this manner, we can plot a curve of twisting moments on a crank angle base for the after arm of each cylinder. They are shown by the dotted curves in Figs. 39 to 42 (Plate XIV.). The convention as regards sign is that explained in § 6, and the angle in all cases is the angle turned through by the crank considered, from its bottom dead centre.

(2) *Bending Moments in Plane of the Arms.*

§ 27. Next, the bending moment on the after arm, in the longitudinal plane of the arm and shaft, is the same at every section, and is equal to (Fig. 2, Plate XI.)—

$$R_1 . CG - R . AG, \textit{ i.e.}, \text{ to } R_2 . HG \text{ or } R . \frac{AC . GH}{CH},$$

where R is the component of the force in the rod resolved along the crank radius. This bending moment is the same multiple of R that the twisting moment on the arm is of Q, and, therefore, for the high-pressure cylinder, is equal to ·865 R. Thus, when the H.P. piston has moved through one-quarter of the up-stroke, the effective force on the piston is 29·6 tons, and, therefore, the force transmitted through the rod is 30·5 tons. In this position the component R along the radius is 17·9 tons, and the bending moment in this position is consequently $17{\cdot}9 \times {\cdot}865 = 15{\cdot}5$ foot tons. By proceeding in this way, we can draw a curve of bending moments on the after arm, in the longitudinal plane of the arm and shaft, for each cylinder. They are shown in Figs. 39 to 42 (Plate XIV.) by the chain-dot curves.

(3) *Bending Moments Perpendicular to Plane of Arms.*

§ 28. The bending moment on the after arm in the plane perpendicular to the longitudinal plane of the arm and shaft has been discussed in § 6. It varies uniformly from the magnitude of the twisting moment on the crank pin, at the crank pin end, to the magnitude of the twisting moment on the crank shaft at the crank shaft end. Thus the bending moment at each end of the arm can be readily obtained from our previous curves. Usually the maximum bending moment takes place at the crank shaft end. The determination of the bending stress at this section is, however, a difficult matter, owing to the connection of the arm with the shaft. For this reason the bending moment on the arm midway between the pin and shaft is the one that has been considered, because, at this mid-section, the section is given by the dimensions of the arm. It is clearly equal to the arithmetical mean of the bending moments at the

TABLE V.

MOMENTS ON AFTER ARM, IN FOOT TONS.

Cylinder.	F.L.P.	H.P.	M.P.	A.L.P.
Maximum twisting moment ...	40·0	60·0	64·0	39·0
Maximum bending moment in plane of cranks	42·0	52·0	58·0	45·0
Maximum bending moment perpendicular to plane of cranks ...	66·0	169·0	222·0	225·0

two ends of the arm. Thus, for example, in the after arm of the H.P. cylinder, when the crank arm has turned through 66° from the bottom dead centre, the twisting moment on the crank pin (from Fig. 32, Plate XIII.) is 94·2 foot tons, and on the after journal (from Fig. 24) 119·4 foot tons (*i.e.* equal to 70·0 + 49·4), so that the bending moment on the after arm in the plane perpendicular to the longitudinal plane of the arm and shaft, at the section midway between the pin and shaft, is 106·8 foot tons. By proceeding in this way, the full lined curves in Figs. 39 to 42 (Plate XIV.) have been drawn. The ordinates to them give the bending moments on the mid-section of the after arm, for the different cylinders in different positions of the crank. The principal results are given in Table V.

Longitudinal Curve of Moments for Shaft (Neglecting Effect of Continuity).

§ 29. Thus, neglecting the effect of continuity of the shaft, the preceding curves give us the straining actions at any particular section during one complete revolution of the shaft. It is interesting to see how the straining actions for some particular position of the crank shaft vary from section to section as we proceed from forward to aft. A "longitudinal" curve of moments for the crank shaft and pins is given in Fig. 43 (Plate XIV.). The position of the shaft taken corresponds to the position of the H.P. crank arm for which the equivalent twisting moment on the H.P. pin was a maximum, that is, when the H.P. crank has turned through an angle of 290° from its bottom dead centre. An end view, looking from aft in a forward direction, is shown in Fig. 44. On a longitudinal axis as base, ordinates have been erected at every section giving the values of the direct bending moment (M), direct twisting moment (T), values of—

$$\sqrt{M^2 + T^2} \quad \text{and of} \quad M + \sqrt{M^2 + T^2},$$

and the results obtained are shown in Fig. 43. The curve of M is shown by the faint dotted line. It is readily plotted, because we have already got the bending moment at the centre section of the crank pin and the diagram between any two journals is a triangle. Thus for the H.P. crank, the angle is 290°, and the bending moment on the pin is, from Fig. 28 (Plate XIII.), 129 foot tons; for the M.P., the angle is 36° and the bending moment on the pin, from Fig. 29, is 56 foot tons. The central part of the shaft is not subjected to bending. The curve of T is shown by the stepped chain-dot line, and, again, is at once plotted from our previous curves. Thus, the twisting moment on the H.P. pin, for an angle of 290°, is, from Fig. 32, 136 foot tons; the twisting moment on the aft journal of the H.P. cylinder is, from Fig. 24 (remembering the F.L.P. crank is at an angle of 125° from its bottom dead centre), 199 foot tons, and so on. Having obtained these curves, the curves of—

$$\sqrt{M^2 + T^2} \quad \text{and} \quad M + \sqrt{M^2 + T^2}$$

are immediately deduced from them, and are represented by the thick dotted and full lined curves respectively. The maximum height of the curve of—

$$M + \sqrt{M^2 + T^2}$$

is at the H.P. pin, and is equal to 316 foot tons as before. In the position of the shaft taken, it will be noticed that the twisting moment on the shaft is a maximum at the after journal of the M.P. cylinder, the reason being that the A.L.P. crank has just passed its top dead centre, and, as already pointed out in § 19, is being driven by, instead of driving, the crank shaft.

Effect of Continuity of the Shaft.

§ 30. The only remaining thing to consider, so far as the straining actions are concerned, is the effect of continuity of the shaft. The method of procedure has already been given in §§ 11, 12, 13. In order to obtain numerical data for comparison with the preceding results, it will be sufficient to consider fully the effect of continuity in one or two positions of the crank shaft. The first position chosen is that for which, neglecting the effect of continuity, the equivalent twisting moment on the H.P. pin is a maximum, viz., when the H.P. crank has turned through an angle of 290° from its bottom dead centre. Figs. 43 and 44 (Plate XIV.) refer to this position of the shaft, whilst Fig. 11 (Plate XI.) represents an isometric projection of it. Following the general method previously given, we must first resolve the force acting along each rod into two components parallel to two perpendicular planes, and then apply the equation of three moments, given in § 11, to these two separate sets of forces. The planes chosen are the longitudinal plane of the engine (in this case a vertical plane) and the plane through the axis of the shaft perpendicular to it. Following the convention in § 13, forces acting downwards, in the longitudinal plane, are reckoned positive, and upwards, negative. In the perpendicular plane, the positive forces, looked at from the after end, act from right to left. The forces (V) in the longitudinal plane are simply the effective piston forces, whilst the forces (W) perpendicular to the plane are the guide reactions. The forces acting are given in Table VI.

TABLE VI.

Cylinder.	F.L.P.	H.P.	M.P.	A.L.P.
Angle of crank with bottom centre	125°	290°	36°	194°
Force in longitudinal plane, *i.e.*, piston pressure	$-39{\cdot}4\ (V_1)$	$+76{\cdot}0\ (V_2)$	$-33{\cdot}5\ (V_4)$	$-15{\cdot}5\ (V_5)$
Force perpendicular to longitudinal plane, *i.e.*, guide reaction ...	$-\ 8{\cdot}5\ (W_1)$	$-18{\cdot}0\ (W_2)$	$-\ 5{\cdot}0\ (W_4)$	$+\ 1{\cdot}0\ (W_5)$

The dimensions of the shaft are given in Fig. 15 (Plate XII.), so that if the five spans be denoted by suffixes 1, 2, 3, 4, 5, we have—referring to the symbols used in § 11—

$$l_1 = 5{\cdot}94\ ;\quad l_2 = 6{\cdot}55\ ;\quad l_3 = 7{\cdot}9\ ;\quad l_4 = 6{\cdot}55\ ;\quad l_5 = 5{\cdot}94\ ;$$
$$k_1 = \ {\cdot}495\ ;\ k_2 = \ {\cdot}51\ ;\quad k_3 = \ {\cdot}0\ ;\quad k_4 = \ {\cdot}49\ ;\quad k_5 = {\cdot}505.$$

Considering, therefore, the V forces, the equation of three moments (Equation (2), § 11) for the spans (1) and (2) gives—

$$5{\cdot}94\ M_a + 24{\cdot}98\ M_b + 6{\cdot}55\ M_c = 13{\cdot}17\ V_1 + 15{\cdot}93\ V_2\ ;$$

for the spans (2) and (3)—

$$6{\cdot}55\ M_b + 28{\cdot}9\ M_c + 7{\cdot}9\ M_d = 16{\cdot}16\ V_2\ ;$$

for the spans (3) and (4)—

$$7{\cdot}9\ M_c + 28{\cdot}9\ M_d + 6{\cdot}55\ M_e = 16{\cdot}16\ V_4\ ;$$

and for the spans (4) and (5)—

$$6{\cdot}55\ M_d + 24{\cdot}98\ M_e + 5{\cdot}94\ M_f = 15{\cdot}93\ V_4 + 13{\cdot}17\ V_5.$$

In addition we assume that—

$$M_a = 0 \quad \text{and} \quad M_f = 0,$$

and so, for the shaft considered, get the equations—

$$\begin{aligned}
M_a &= 0,\\
M_b &= {\cdot}5623\ V_1 + {\cdot}5122\ V_2 + {\cdot}0366\ V_4 - {\cdot}01056\ V_5,\\
M_c &= -\ {\cdot}1388\ V_1 + {\cdot}48\ V_2 - {\cdot}1397\ V_4 + {\cdot}0404\ V_5,\\
M_d &= {\cdot}0404\ V_1 - {\cdot}1397\ V_2 + {\cdot}48\ V_4 - {\cdot}1388\ V_5,\\
M_e &= -\ {\cdot}01056\ V_1 + {\cdot}0366\ V_2 + {\cdot}5122\ V_4 + {\cdot}5623\ V_5,\\
M_f &= 0.
\end{aligned}$$

These equations apply equally to our second plane of reference, provided that for V we substitute W.

Considering the V forces, we at once get by substitution in these equations—

$$M_a = 0, \quad M_b = +\ 15{\cdot}8, \quad M_c = +\ 46{\cdot}0, \quad M_d = -\ 26{\cdot}1, \quad M_e = -\ 19{\cdot}0, \quad M_f = 0,$$

which give us the bending moments at the journals in the longitudinal plane. The shearing force is then obtained from equation (3) in § 11, care being taken regarding the sign. Thus, for example, our five equations will be—

$$\begin{aligned}
+\ 15{\cdot}80 &= 0 + F_a \times 5{\cdot}94 - 39{\cdot}4 \times 3{\cdot}0\\
+\ 46{\cdot}00 &= +\ 15{\cdot}8 + F_b \times 6{\cdot}55 + 76 \times 3{\cdot}22\\
-\ 26{\cdot}1 &= +\ 46{\cdot}0 + F \times 7{\cdot}9\\
-\ 19{\cdot}0 &= -\ 26{\cdot}1 + F_d \times 6{\cdot}55 - 33{\cdot}5 \times 3{\cdot}33\\
0 &= -\ 19{\cdot}0 + F_e \times 5{\cdot}94 - 15{\cdot}5 \times 2{\cdot}94,
\end{aligned}$$

whence

$$F_a = +\ 22{\cdot}6, \quad F_b = -\ 32{\cdot}8, \quad F_c = -\ 9{\cdot}15, \quad F_d = +\ 18{\cdot}1, \quad F_e = +\ 10{\cdot}9, \quad F_f = 0.$$

These are the shearing forces immediately to the right of the different journals.* By

* The reactions at the journals are + 22·6, − 16·0, − 52·35, + 27·25, + 26·28 and + 4·62 respectively. In Fig. 11 (Plate XI.), the crank pin forces and reactions have been drawn in the proper direction, as shown by the arrows.

proceeding in the same way, we get, in the plane perpendicular to the longitudinal plane, the results—

$$M_a = 0, \quad M_b = -14{\cdot}2, \quad M_c = -6{\cdot}72, \quad M_d = -{\cdot}37, \quad M_e = -2{\cdot}57, \quad M_f = 0.$$

$$F_a = +1{\cdot}9, \quad F_b = +10{\cdot}0, \quad F_c = +{\cdot}8, \quad F_d = +2{\cdot}2, \quad F_e = -{\cdot}06, \quad F_f = 0.^*$$

The second system of forces is, of course, of very much less importance than the first system. We are now in a position to draw longitudinal curves of bending moments for the shaft and pins in the two planes of reference. Neglecting the effect of continuity, the curve of bending moments in the longitudinal plane of the cylinders would be represented by the chain-dotted line in Fig. 45 (Plate XIV.). Thus, for example, the height of the triangle for the H.P. cylinder is merely

$$\frac{76{\cdot}0 \times 3{\cdot}33 \times 3{\cdot}22}{6{\cdot}55} = 124{\cdot}2,$$

and, according to the convention as regards signs, must be reckoned negative. The effect of continuity is taken into account by erecting ordinates at the different journals equal to the bending moments as just calculated, and so obtaining the dotted lines in Fig. 45. The bending moment when continuity is considered is then obtained by taking the algebraic sum of the ordinates to those two curves, and so obtaining the full lined curve. The bending moment curve in the second plane must be obtained in the same manner. It is shown in Fig. 46, in which, as in Fig. 45, the chain-dotted line refers to the bending moment neglecting continuity, the dotted line curve to the bending moment curve due to continuity, and the full line curve the net bending moment when continuity is considered. To get the resultant bending moment (M) at any section, we must take corresponding ordinates to the full lined curves in Figs. 45 and 46, square them, add them together, and take the square root. If we do this, we get as our curve of resultant bending moments the faint dotted curve in Fig. 47. Thus, for example, at the H.P. pin, the bending moment in the longitudinal plane is 93 foot tons and perpendicular to it 19 foot tons, so that the resultant bending moment is

$$\sqrt{93^2 + 19^2} = 95{\cdot}0 \text{ foot tons.}$$

§ 31. Next consider the twisting moments. The twisting moments on the different parts of the shaft are, of course, the same whether we consider the shaft to be continuous or not, and therefore are already known. The twisting moment on any pin has been shown to be (§ 13) equal to the twisting moment transmitted from the cylinders forward of that pin, together with the moment of the shearing forces, in the two reference planes, about the crank pin considered, the shearing forces being taken immediately aft of the forward journal of the cylinder considered. Thus, for example, for the

---

* The reactions at the journals are + 1·9, + 16·6, + 8·8, + 1·4, + 2·74 and − ·94 respectively.

F.L.P. pin, the shearing forces in and perpendicular to the longitudinal plane are + 22·6 and + 1·9, and, therefore, referring to Fig. 44, the moment of these shearing forces (reckoned clockwise) is $22{\cdot}6 \times 1{\cdot}63 + 1{\cdot}9 \times 1{\cdot}14 = +\ 39{\cdot}0$. Since there is no twisting moment transmitted forward of the F.L.P. cylinder, this represents the total twisting moment on the F.L.P. pin. Or again, the shearing forces aft of the forward journal of the H.P. cylinder are − 32·8 and + 10·0, so that the clockwise twist on the H.P. pin is $32{\cdot}8 \times 1{\cdot}86 - 10{\cdot}0 \times {\cdot}7 = 54{\cdot}0$. The twist from the F.L.P. cylinder in this position is 72·0, so that the total twist in the H.P. pin is 126·0 foot tons. In the same way, the twisting moment on the M.P. pin due to the shearing forces is 18·2, and on the A.L.P. pin − 5·6, whilst the total twist on the M.P. pin is 218·0, and on the A.L.P. pin is 224·0 foot tons. The curve of twisting moment T, on the axis of the shaft as base, is

TABLE VII.

MOMENTS IN FOOT TONS ON JOURNALS AND PINS WHEN THE F.L.P. CRANK HAS TURNED THROUGH 125° FROM THE BOTTOM DEAD CENTRE.

	The different sections of the Shaft considered as separate beams				The Shaft considered as a continuous beam.			
	M	T	$\sqrt{M^2+T^2}$	$M+\sqrt{M^2+T^2}$	M	T	$\sqrt{M^2+T^2}$	$M+\sqrt{M^2+T^2}$
Journal	0	0	0	0	0	0	0	0
F.L.P. Pin ...	58	37·7	70·9	131	66·7	39	77·4	144
Journal	0	72	72	72	21·2	72	75	96
H.P. Pin ...	129	136	187	316	95·3	126	158	253
Journal	0	199	199	199	46·6	199	205	252
Journal	0	199	199	199	26·1	199	202	228
M.P. Pin ...	55	216	223	278	31	218	219	250
Journal	0	230	230	230	19·2	230	231	250
A.L.P. Pin ...	22·6	225	226	249	11·8	224	225	237
Journal	0	220	220	220	0	220	220	220

shown by the chain-dot line in Fig. 47 (Plate XIV.). Having thus got the curves of M and T, we can plot the curves of—

$$\sqrt{M^2+T^2} \quad \text{and} \quad M+\sqrt{M^2+T^2},$$

as we did in Fig. 43. The curve of

$$\sqrt{M^2 + T^2}$$

is represented by the thick dotted line, and that of

$$M + \sqrt{M^2 + T^2}$$

by the full line. A comparison of Figs. 43 and 47 at once shows, for the crank position considered, the effect of continuity. Clearly, the effect of continuity is to make the moments more uniform along the shaft. The equivalent twisting moment on the H.P. pin is reduced from 316 to 253 (*i.e.*, 20 per cent.), that on the M.P. pin from 278 to 250 (*i.e.*, 10 per cent.), and on the A.L.P. pin from 249·0 to 237 (*i.e.*, 5 per cent.). That on the F.L.P. pin is increased from 131·0 to 144 (*i.e.*, 10 per cent.). Thus the maximum reduction, viz., that on the H.P. pin, is a matter of 20 per cent. The equivalent twisting moment on all the journals is, of course, increased. That on the after journal of the H.P. cylinder is increased from 199 to 252, an increase of 25 per cent. This

TABLE VIII.

MOMENTS IN FOOT TONS WHEN F.L.P. CRANK HAS TURNED THROUGH 330° FROM ITS BOTTOM CENTRE.

	The different sections of the Shaft considered as separate beams.				The Shaft considered as a continuous beam.			
	M	T	$\sqrt{M^2+T^2}$	$M+\sqrt{M^2+T^2}$	M	T	$\sqrt{M^2+T^2}$	$M+\sqrt{M^2+T^2}$
Journal	0	0	0	0	0	0	0	0
F.L.P. Pin ...	74·5	19	77	151	72·3	21·6	75·4	148
Journal	0	38·6	38·6	38·6	9·1	38·6	39·7	49
H.P. Pin ..	87·3	81·7	119	207	68·3	74·2	101	169
Journal	0	126	126	126	39·2	126	132	172
Journal	0	126	126	126	33·5	126	131	164
M.P. Pin ...	87	174	194	281	54	178	186	240
Journal	0	224	224	224	22·5	224	225	247
A.L.P. Pin ...	6	225	225	231	15·7	219	220	236
Journal	0	226	226	226	0	226	226	226

moment is practically the same as on the H.P. and M.P. pins, and on the after journal of the M.P. cylinder. The principal results are expressed in Table VII.

§ 32. The effect of continuity in a second position has also been worked out, viz., that position of the crank shaft for which the combined turning moment of the four cylinders is a maximum. That happens when the F.L.P. crank has turned through an angle of 330° from its bottom dead centre. The crank shaft, therefore, has completed rather more than half a turn from our previous position. The principal results are given in Table VIII. on the preceding page. For this position, the equivalent twisting moments on the F.L.P. and A.L.P. pins are little altered, but that on the H.P. pin is reduced from 207 to 169 (*i.e.*, 18 per cent.), and on the M.P. pin from 281 to 240 (*i.e.*, 15 per cent.). The equivalent twisting moments on the H.P. after journal and the M.P. forward journal are each increased about 36 per cent.

The bending and twisting moments on the crank arms need not be discussed, for the reason which is given in § 38.

TABLE IX.

STRAINING MOMENTS IN FOOT TONS ON THE ASSUMPTION THAT THE SHAFT IS EFFECTUALLY CONSTRAINED IN DIRECTION AT EACH JOURNAL.

	Position of Crank Shaft corresponding to Table VII.				Position of Crank Shaft corresponding to Table VIII.			
	M	T	$\sqrt{M^2+T^2}$	$M+\sqrt{M^2+T^2}$	M	T	$\sqrt{M^2+T^2}$	$M+\sqrt{M^2+T^2}$
Journal ...	29·0	0	29·0	29·0	37·2	0	37·2	37·2
F.L.P. Pin	29·0	37·7	47·6	73·6	37·2	19·0	41·8	79·0
Journal ...	29·0 64·5	72·	78·0 97·0	107· 162·	37·2 43·6	38·6	53·6 58·2	91· 102·
H.P. Pin .	64·5	136·	151·	215·	43·6	81·7	92·6	136·
Journal ...	64·5	199·	209·	274·	43·6	126·	133·	177·
Journal ...	27·5	199·	201·	229·	43·5	126	133·	177·
M.P. Pin .	27·5	216·	218·	245·	43·5	174·	180·	224·
Journal ...	27·5 11·6	230·	232· 231·	260· 243·	43·5 3·0	224·	228· 224·	272· 227·
A.L.P. Pin	11·6	225·	226·	238·	3·0	225·	225·	228·
Journal ...	11·6	220·	221·	233·	3·0	226·	229·	229·

Results when the Shaft is Effectually Constrained at each Bearing.

§ 33. It will be interesting to compare these results with those obtained on the assumption that the shaft is effectually constrained in direction at the journals (§ 14). To do so, we will assume that the force along each rod acts in the plane midway between the journals, an assumption which is practically true. In that case, taking the span from centre to centre of the bearings (as we do when we assume the shaft to be merely supported at the journals) the bending moments on the pins will be simply half of those estimated on the first assumption, and this will likewise be the bending moment on each journal. The reactions will be the same as before, and therefore the twisting moments on the pin and journals are unaltered. We thus get Table IX., which ought to be compared with Tables VII. and VIII.

§ 34. But a correction is required in Table IX. If we assume the bearings to act simply as points of support, we must take, as the span, the distance from centre to centre of the journals, as we have done throughout. But when we assume that the shaft is effectually constrained in direction by the bearings, then, strictly speaking, we must take as the span, not the distance from centre to centre of the journals, but the distance between the inner ends of the bearings. This would reduce the span by about 2·5 feet in all cases (see Fig. 12, Plate XII.), so that the ratio of the spans in the two cases would be ·58 for the F.L.P. and A.L.P. cylinders and ·62 for the H.P. and M.P. cylinders. Since the bending moment is proportional to the span, the values of M in the last table would have to be multiplied by ·58 for the F.L.P. and A.L.P. cylinders, and by ·62 for the H.P. and M.P. cylinders. The twisting moments on the pins and shaft would of course remain unaltered, so that the necessary corrections could be at once made. Thus, for the H.P. pin, taking the first position of the crank shaft, we should get

$$M = 40, \quad T = 136, \quad \sqrt{M^2 + T^2} = 142, \quad M + \sqrt{M^2 + T^2} = 182,$$

as against 215 in Table IX. and 253 in Table VII. On the H.P. after journal we should get

$$M = 40, \quad T = 199, \quad \sqrt{M^2 + T^2} = 201, \quad \text{and} \quad M + \sqrt{M^2 + T^2} = 241,$$

as against 274 in Table IX. and 252 in Table VII. If we compare this last estimate with that in Table VII. (viz., 252) we see that it is actually less than our previous estimate, when the effect of continuity is considered, even for the journal. For the crank pin, it is very much less. It would appear, therefore, to be undesirable to take the length of the span as being the distance between the inner ends of the bearings, because we might then under-estimate the true value. If we take the span from centre to centre of the journals, a comparison of Table IX. with Tables VII. and VIII. shows that, so far as the journals are concerned, the results in Table IX. agree fairly well with those in Tables VII. and VIII., when the continuity of the shaft is considered, being, in all cases, somewhat in excess of the results given in Tables VII. and VIII. The equivalent twisting moments on the pins as given in Table IX. are, in all cases, less than the equivalent twisting moments as given in Tables VII. and VIII. when the effect of continuity is considered.

Working Assumptions.

§ 35. It would therefore appear desirable, in estimating the straining actions on the pins, to assume that the shaft is not constrained in any manner at the journals, so that, for the pins, the equivalent twisting moments are given in Figs. 35 to 38 (Plate XIII.). Also, in estimating the straining actions on the journals, it is desirable to assume that the bearings effectually constrain the journals—the span being taken from centre to centre of adjacent journals. In that case, the curve of direct bending moment on each of the two journals corresponding to any pin is exactly similar to the curve of bending moments given in Figs. 27 to 30 (Plate XIII.), but (very approximately) is only half the height of these curves. The twisting moment on any journal is the same as before, and is, therefore, given by Figs. 24, 25, 26.

Thus we can at once get the values of—

$$\sqrt{M^2 + T^2} \text{ and } M + \sqrt{M^2 + T^2}$$

on this basis for each journal. Thus, for example, taking the forward journal of the H.P. cylinder, when the H.P. crank has turned through, say, 66° from the bottom dead centre, the bending moment on the F.L.P. pin is (from Fig. 27) since the F.L.P. crank has turned through 261° from the bottom dead centre, 55·0, and on the H.P. pin (from Fig. 28) 50·0, on the assumption that the bearings exercise no constraint; whilst the twisting moment transmitted from the F.L.P. cylinder is (from Fig. 24) 70 foot tons. On the assumption that the bearings effectually constrain the shaft, the bending moment on the journal considered as the F.L.P. aft journal will be 27·5, and, considered as the H.P. forward journal, will be 25·0. Thus the two values of

$$\sqrt{M^2 + T^2}$$

will be 75·3 and 74·4, and of

$$M + \sqrt{M^2 + T^2}$$

will be 102·8 and 99·4, according as we consider the bending moment on the journal as being due to the F.L.P. or H.P. cranks respectively.

In this way we can draw curves giving the values of

$$M + \sqrt{M^2 + T^2}$$

on the different journals on the assumption that the journals are effectually constrained by the bearings. They are shown in Figs. 48 to 51 (Plate XV.), which ought to be compared with Figs. 24, 25, 26 (Plate XIII.), which give the corresponding quantities on the assumption that the bearings exercise no constraint on the journals. The faint curve in Fig. 51 refers to the F.L.P. forward journal. The faint curve in Fig. 48 refers to the aft journal of the F.L.P. cylinder; that is to say, when the bending moment on that journal is taken to be due to the force on the F.L.P. crank; the full curve is the curve of equivalent twisting moment on the same journal when the bending moment is taken to be due to the force on the H.P. crank. Thus, for an angle of 261°, the ordinates to these two curves are 102·8 and 99·4 respectively. Similarly for Figs. 49 and 50, that curve is thickened which gives the maximum value of

$$M + \sqrt{M^2 + T^2}$$

The thick curve of Fig. 51 represents the value of

$$M + \sqrt{M^2 + T^2}$$

for the after journal of the A.L.P. cylinder. From these curves we obtain Table X., which gives the maximum moment on the different journals. For comparison the maximum values are also given as obtained from Figs. 24, 25, 26.

TABLE X.

MAXIMUM VALUES OF THE STRAINING MOMENTS IN FOOT TONS ON THE DIFFERENT JOURNALS.

	Assuming no constraint on bearings.								Assuming the bearings effectually constrain the Shaft.							
	F.L.P.		H.P.		M.P.		A.L.P.		F.L.P.		H.P.		M.P.		A.L.P.	
	fore.	aft.	fore.	aft.	fore.	aft.	fore.	aft.	fore.	aft.	fore.	aft.	fore.	aft.	fore.	aft.
Maximum twisting moment	0	87·6	87·6	202	202	230	230	226	0	87·6	87·6	202	202	230	230	226
Maximum bending moment	0	0	0	0	0	0	0	0	39·5	39·5	64·5	64·5	66	66	40	40
Maximum value of— $\sqrt{M^2 + T^2}$	0	87·6	87·6	202	202	230	230	226	39·5	95·0	98	210	205	229	230	227
Maximum value of— $M + \sqrt{M^2 + T^2}$	0	87·6	87·6	202	202	230	230	226	39·5	133	163	274	233	271	254	262

The most severely strained journal is therefore the aft journal of the H.P. cylinder. The second assumption gives a maximum equivalent twist of 274 foot tons as against 202 on the first assumption—an increase of about 36 per cent.

Maximum Stresses induced in the Different Parts of the Crank Shaft.

§ 36. It will be interesting to collect the maximum moments on the different journals, pins and arms, and, knowing the dimensions of the different parts, to estimate the intensities of stress set up on the assumption that these moments are gradually applied. The values of the maximum straining actions on the different journals, pins, and arms are given in the preceding tables, and are reproduced in Tables XI. and XII.

The ratio of the maximum equivalent twisting moment on the different journals and pins to that of the mean twisting moment on the propeller shaft ought to be particularly noticed. That ratio is a maximum for the H.P. pin, and is then 1·63.

The shaft and the pins are hollow, the internal and external diameters of the shaft (journals) being 10 and 19 inches respectively, and of the pins 12 and 21 inches (see Fig. 12). The intensity of stress f, due to a twisting moment T, is given by the formula—

$$f = T \cdot \frac{16\, d_1}{\pi\,(d_1^4 - d_2^4)},$$

where d_1 and d_2 are the external and internal diameters respectively. If T be in foot tons, d_1 and d_2 must be expressed in feet, and then f will be in tons per square foot. Thus, substituting, we find that for the journals

Stress in tons per square foot = 1·39 T,

or—

Stress in tons per square inch = ·00965 T ;

and for the pins—

Stress in tons per square foot = 1·07 T,

or—

Stress in tons per square inch = ·00742 T.

When dealing with bending moments, these co-efficients must be doubled. Thus, in all our previous curves, we can merely scale off the value of T or M in foot tons, and multiply the reading so obtained by the proper co-efficient, in order to get the intensity of stress induced. The maximum stresses induced in the journals are given in Table X. It will thus be seen that, assuming the loads to be applied gradually,* the maximum stress on the journals is practically $2\frac{2}{3}$ tons per square inch, and on the pins $2\frac{1}{3}$ tons per square inch. When the effect of continuity of the shaft is considered, the stress on the pins and journals would be found to be generally less than these quantities, and the two cases worked out in §§ 30, 31, 32, give us an idea of the magnitude of the decrease.

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* In this connection it must be remembered that, if a load be suddenly applied to an unstressed structure, the stress induced is double that which would be induced if the load were applied gradually. If a load on a structure be suddenly reversed in direction, the stress induced is trebled.

The tunnel shaft is 10 inches internal and 17½ inches external diameter, so that, assuming it to be subjected only to twisting, the maximum stress in it is 2·89 tons per square inch.

TABLE XI.

MAXIMUM STRAINING MOMENTS, IN FOOT TONS, AND THE CORRESPONDING STRESSES INDUCED IN TONS PER SQUARE INCH.

	AFTER JOURNALS. Assuming that the bearings effectually constrain the shaft (from Table X.).				CRANK PINS. Assuming that the bearings exercise no constraint on the shaft (from Table IV.).			
Cylinder.	F.L.P.	H.P.	M.P.	A.L.P.	F.L.P.	H.P.	M.P.	A.L.P.
Maximum value of twisting moment	87·6	202	230	226	45·0	136	215	225
Maximum value of bending moment	39·5	64·5	66·0	40·0	79·0	129	132	80·0
Maximum value of $\sqrt{M^2 + T^2}$ ...	95·0	210	229	227	89·0	187	222	226
Maximum value of $M + \sqrt{M^2 + T^2}$ (equivalent twisting moment)...	133	274	271	262	167	316	283	300
Ratio of max. equivalent twisting moment to mean twisting moment on propeller shaft (194·2)...	·69	1·41	1·40	1·35	·86	1·63	1·46	1·54
Stress due to maximum twist	·85	1·95	2·22	2·18	·33	1·01	1·60	1·67
Stress due to maximum bending	·76	1·24	1·27	·77	1·17	1·91	1·96	1·19
Maximum shear stress (due to $\sqrt{M^2 + T^2}$) above ...	·92	2·02	2·21	2·19	·66	1·39	1·65	1·68
Maximum direct stress (due to $M + \sqrt{M^2 + T^2}$) above	1·28	2·64	2·62	2·53	1·24	2·34	2·10	2·22

§ 37. Next, consider the arms. The section is a rectangle of dimensions 1 ft. by 1·96 ft. In estimating the stress due to the bending moment in the longitudinal plane of the arms, the shorter side must be taken as the "depth" of the section; whilst for the bending moment perpendicular to the plane of the arms, we must take the longer side as the depth of the section. Thus, for these two moments, the stress induced, in tons per square foot, will be given by the formula—

$$f = \frac{6\,M}{b\,d^2},$$

and will therefore be ·306 M and ·156 M; or, in tons per square inch, ·00213 M and ·00109 M respectively, M being in foot tons.

The stress due to the twisting moment cannot be obtained from the ordinary formula for circular shafts, because the section is a rectangle. The maximum shearing stress takes place at the centre of the larger side, and the relation between that maximum stress f and the twisting moment T is, according to St. Venant, given by the formula—

$$f = T \, . \, \frac{3\,l + 1{\cdot}8\,s}{s^2\,l^2},$$

in which s and l are the shorter and longer sides of the rectangle respectively. By substitution, we find that—

f in tons per square foot = T in foot tons × 2·0,

or—

Stress in tons per square inch = ·0139 T.

Applying these formulæ, we thus obtain the results given in Table XII.

TABLE XII.

Maximum Straining Moments, in Foot Tons, and Stresses induced, in Tons per Square Inch, at the Midsection of the After Arm, the Bearings being assumed to exercise no constraint.

Cylinder.	F.L.P.	H.P.	M.P.	A.L.P.
Maximum twisting moments ...	40·0	60·0	64·0	39·0
Maximum bending moment in plane of cranks	42·0	52·0	58·0	45·0
Maximum bending moment perpendicular to plane of cranks ...	66·0	169·0	222·0	225·0
Shear stress due to twisting moment	·56	·83	·89	·54
Direct stress due to bending in plane of cranks	·09	·11	·12	·10
Direct stress due to bending perpendicular to plane of cranks ...	·07	·18	·24	·25

§ 38. Thus the stress due to the twisting moment is of much greater magnitude than that due to bending. Moreover, these values do not take place simultaneously (see Figs. 39 to 42, Plate XIV.). The stresses due to bending are so small as to be negligible, and it is on account of the smallness of these stresses that the straining actions on the arms, when the effect of the continuity of the shaft is considered, have not been discussed. The effect of continuity would be to reduce the stresses.

The stresses due to the shearing force at the different sections will be small relatively to the stresses due to the moments, and need not, therefore, be discussed.

DISCUSSION.

Mr. Macfarlane Gray (Member of Council): Mr. Chairman and Gentlemen, I have not had an opportunity of reading the paper through. What I have read, however, shows me that Professor Dunkerley has done very good service, especially to the younger members of this Institution, by preparing this paper so carefully. The methods of Professor Dunkerley, and also of Professor Dalby, are not exactly on the lines I generally adopt. I go in more for diagrammatic solutions. The theorem of three moments is, I think, clearer, especially in regard to the practical unreliability of its possible cases, when so treated. I think, however, that the paper is an admirable one, from the author's point of view, and well adapted for those who are studying this problem. It is an advantage through life to have been well grounded in any subject at some time or other. The thoroughness tells in dealing with other questions than those immediately referred to. I wish my words to be taken only as praise for the paper.

Professor T. A. Hearson (Member): Mr. Chairman and Gentlemen, I regret that I have not had an opportunity of reading this paper before coming here this evening. You can understand that it is, therefore, somewhat difficult for me to make any remarks in the way of criticism under such circumstances. I join with Mr. Macfarlane Gray in expressing my pleasure in hearing the paper, and particularly with reference to the lucid way in which Professor Dunkerley has submitted it to us, without our having to go through the paper in all its printed detail. The subject is an exceedingly interesting example of the problem of crank shaft strains, and it will be very useful for the students of the Institution and others, as indicating a method to be followed. With respect to Mr. Macfarlane Gray's remark as to the way in which the results are worked out, referring particularly to the equation of three moments, I think that if one could find some way of getting the results other than by that method, we should be very glad to know of it. The supposition in which the shaft is free to incline at the bearings, and is not constrained by connection to adjacent lengths, will lend itself to the draughtsman's method; but when you have the constrained condition of a continuous shaft in many bearings, in which you have to take account of the elasticity, I do not see how to get at results except by the well-known method of the problem of three moments. The theory of the three moments may be extended to the case in which the bearings are put out of line by deformation in the vessel, and I am afraid that that is a condition to which the shafts are subjected when in actual work. When we see, as the result of this calculation, that the greatest stress on the material of the shaft is less than three tons on the square inch, and when we know that such shafts as these do break, we see that the suppositions on which the results have been worked out are not the worst to occur in practical circumstances. These latter are probably not capable of being dealt with by theoretical considerations. They arise when the vessel becomes strained, when the bearings are put out of line, and extra strains are brought into consideration. There is one point more: Professor Dunkerley has pointed out that the maximum equivalent twisting moment is greater forward than it is at the place where one generally expects it, and where one has generally found it to be the greatest. I think that must be due to rather special circumstances. If you look at the diagram of simple bending moment due to the local pressure on the piston between the two supporting bearings, independently of the action on other parts of the shaft, you will see that, for the after low-pressure engine, the magnitude is very much less than it is for the forward low-pressure. I think that must be due to some special local circumstances, that is, the steam does not get into the two cylinders in corresponding quantities, so as to produce equal bending moments. It is fortunate that it is so, because it produces relief at the place where it is advantageous that relief should be provided. But I think that must be more or less incidental to the particular example, and not generally the case. I think that, in general, the maximum equivalent twisting moment is greater towards the after end of the shaft, and not at the place where a high-pressure cylinder is situated.

Mr. A. R. Emdin, Chief Engineer, R.N. (Visitor): Mr. Chairman and Gentlemen, having been asked to take part in this discussion, it might be of interest if, in the first place, I explained, in some way, how the paper came to be written. It happened that, in the position I now hold, I had to propose certain dimensions for the machinery specifications for the Admiralty; and, in searching for some way to reduce the weight of this machinery, it was suggested that the dimensions of the forward crank-pins and crank-shafting might be appreciably reduced from those of the corresponding after parts. In order to make sure that this could be done, a more or less tentative investigation was made into those stresses on the crank-shaft which Professor Dunkerley has explained so thoroughly this evening. The result was practically the same as that which he has just now pointed out at some length; namely, that the forward parts of the crank-shaft were stressed as much, if not more, than the after parts; and the H.P. crank-pin had the greatest stress. This result was quite different to what I expected from my previous ideas on this subject; and, as it is also part of my present duties to lecture on Marine Engine Design to the engineers at the Royal Naval College, I wanted to make quite sure of this point before preparing my notes for the lectures on the subject of crank-shafts. Accordingly, I consulted my friend, Professor Dunkerley, in order that what we told the same people at different times might agree as far as possible. I think that the results I had obtained somewhat surprised him also; and he then undertook to investigate the matter at greater length. The result of his investigations you have had placed before you this evening, and I am sure you will agree that it is a most lucid and convincing exposition of the question. My calculations were made on the assumption that the bearings exercised no restraint on the crank-shaft—that is, each portion of the crank-shaft was considered as independent of the others. I also assumed that only the direct load on the piston was producing the bending moments on the crank-pin; and, after the first discussion with Professor Dunkerley, I worked out some other cases of crank-shafts of engines of a similar character to that described in the paper. In each case, I found results which in every way confirm those of Professor Dunkerley; that is to say, the stress on the H.P. crank-pin was larger than that on any other crank-pin, and although my estimates were not made so carefully or so fully as his, the actual ratios of the maximum stress to the mean stress, in the cases I worked out, also agreed very well. These results I have tabulated as follows; but in order not to trouble you with all the figures, it may be stated that, in the extreme cases, the ratio ranged from $1\frac{1}{2}$ to $2\frac{1}{4}$.

Ratio of Maximum Stress to Mean Stress at Various Parts of a Crank Shaft for a Four-Cylinder Engine.

	Shaft.	A. L. P. Crank Pin.	Shaft.	M. P. Crank Pin.	Shaft.	H. P. Crank Pin.	Shaft.	F. L. P. Crank Pin.	Shaft.
As worked out by Mr. Emdin—									
Number 1	1·19	1·44	1·24	1·4	·93	1·52	·48	·86	0
„ 2	1·14	1·67	1·17	1·73	·87	1·84	·51	1·20	0
„ 3	1·19	1·53	1·19	1·58	1·03	1·72	·46	·94	0
„ 4	1·26	1·30	·72	1·64	·63	1·60	·48	·98	0
„ 5	1·17	1·82	·65	2·20	·93	2·27	·39	1·37	0
„ 6	1·2	1·57	·75	1·74	·65	1·98	·51	1·18	0
As worked out by Prof. Dunkerley—									
As per Tables III. and XI. ...	1·16	1·54	1·18	1·46	1·04	1·63	·45	·86	0
„ Table XI.	1·35	1·54	1·40	1·46	1·41	1·63	·69	·86	—

Professor Dunkerley: The ratio to the mean twisting moment?

Mr. Emdin: Yes, the ratio of the maximum to the mean. There is also another respect in which my figures confirmed what the author has shown in his paper, viz., that the greatest stress occurred when the load on the piston was the greatest; and as these six cases were selected at random, I think I am justified in agreeing with the results given by Professor Dunkerley for the one he has worked out. Of course, as has been pointed out, it must be borne in mind that all these investigations refer to the particular type of engine in which the low-pressure cylinders are placed at the extreme end of the crank-shaft; and each low-pressure cylinder is intended to develop one-half the power of the high or the intermediate. That reservation Professor Dunkerley also clearly makes in his paper, so that the results should be regarded as applicable to that case only. Referring to what Professor Hearson said just now, when he pointed out that the greatest stress on the shaft was only three tons, and yet these shafts break, I should like to take the opportunity to also point out that this discussion refers to shafts made for Admiralty requirements, and which have not been known to break. So that, because, in this case, the stress is limited to three tons per square inch, it is not fair to assume that shafts will break at this comparatively low stress. I think that the results which have been arrived at by the author have justified the Admiralty in not reducing the dimensions of the forward parts of their crank-shafts; and, further, I am sure you will all agree that the paper you have heard has more than justified my consulting Professor Dunkerley, and drawing from him such a lucid and complete investigation of a most important matter.

Mr. T. Donaldson (Associate): Mr. Chairman and Gentlemen, I have been engaged for about a week or ten days in running out the stresses on a shaft, so that perhaps I have the question more vividly before my mind than people who have done it long ago, and I am more in a position to appreciate the value of it, and the immense amount of work that Professor Dunkerley must have put into this paper. I think he is helping us to crystallise our assumptions, and this is a very difficult matter. One must exercise one's judgment, but at the same time it is always a comfort to know that you have a certain amount of precedent to go on. The shaft I have been dealing with has four cranks. I only got Professor Dunkerley's paper about two days ago, and I have not investigated the forward part of the shaft. The after part I take rather in this way:—Between the two bearings

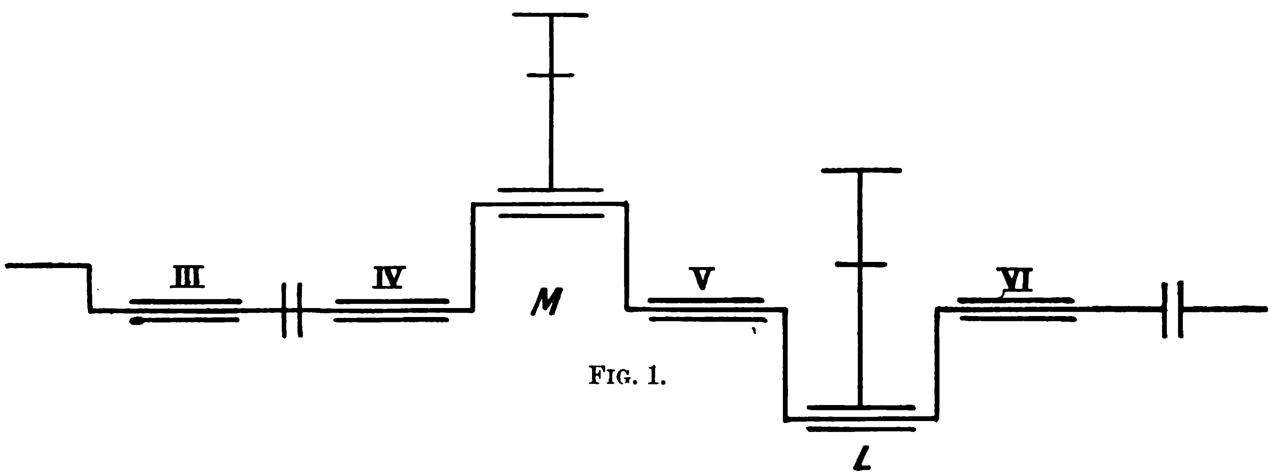

FIG. 1.

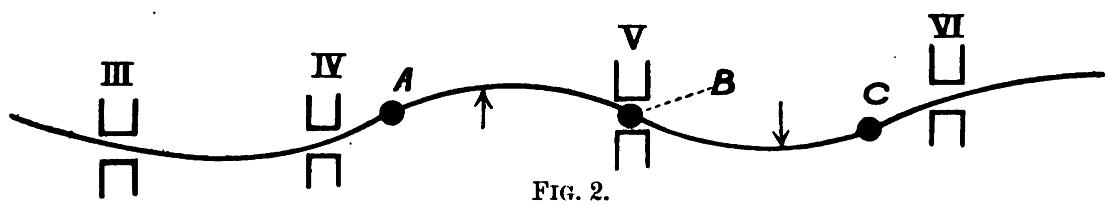

FIG. 2.

III. and IV. (Fig. 1), there would be a slight amount of spring, as indicated in Fig. 2. It seems to me that if the loads on the two pistons (M and L) are approximately equal, there will be a point of inflection somewhere about A, another at B, and another at C, so that the case of those crank pins is not quite so bad as if you took it for the full span. I think if you assume it as simply supported at bearing V. (Fig. 2), and restrained at bearing IV., you get something like $\frac{5}{32}$ W L in the centre of the pin, instead of $\frac{W L}{4}$, but you get $\frac{3}{16}$ W L at bearing IV. I think (though I have not had time to go very carefully into the paper) that we might go rather closer to the wind than Professor Dunkerley has done in most cases in taking the full span. For the journals, you would have to take the shaft, perhaps, as absolutely restrained. You might, as Mr. Seaton, I think, suggests, take the bending moment for the crank-pin as $\frac{1}{6}$ W L, which is a mean between taking the whole span with a bending moment $\frac{1}{4}$ W L, and taking it with two fixed ends, which would give a bending moment $\frac{1}{8}$ W L.

Professor S. DUNKERLEY (Associate): With regard to Mr. Macfarlane Gray's remarks on graphic methods, I think that equation (2), § 11, expresses the theory of continuous girders in a manner which can be readily used by draughtsmen. It appears to me that now, thanks to the spread of technical education within the last ten or fifteen years, the majority of draughtsmen can use a tolerably simple analytical expression as readily as a graphical construction. At the same time, it is hardly necessary to point out to the members of this Institution that there are very few men who possess such a power of vividly emphasising complicated results, and of expressing them by graphical processes, which appeal to all of us, as Mr. Macfarlane Gray; so that I should be glad if he would add, in his printed remarks, an entirely graphical solution of the theory of continuous girders.* With regard to the remarks of Professor Hearson on the equivalent twisting moment on the H.P. pin, namely, the fact that the maximum moment on the A.L.P. pin is less than that on the H.P. pin, I would point out that it is not an accidental case. In Fig. 41, Plate XIV., to which Professor Hearson has alluded, the bending moment on the A.L.P. pin is certainly less than on the H.P. pin, but it must be remembered that the diagram refers, not to maximum values, but to simultaneous values in one position of the shaft. If we compare Figs. 27 and 30, Plate XIII., it will be seen that the maximum bending moments on the two L.P. pins have practically the same value. The only reason (as is pointed out in the paper and to which Mr. Emdin has referred) why the H.P. pin is the most severely stressed pin is, that the H.P. and M.P. cylinders are each developing about double the horse-power of either of the two L.P. cylinders, so that the bending moments on the H.P. and M.P. pins are greater than on the L.P. pins. The cases quoted by Mr. Emdin show that this is not an accidental occurrence in this particular engine, and the results given by him add considerably to any value the paper might possess. In dealing with the bending moments on the shaft and pins, a great many different assumptions might be made, and the remarks of Mr. Donaldson are perfectly justifiable. The three most rational assumptions, to me at least, appear to be those given in the paper; but the results obtained on any other assumption can be readily compared with those there given.

The CHAIRMAN (Mr. A. F. Yarrow, Vice-President): Gentlemen, I want you to authorise me, in the name of the Institution, to offer our very best thanks to Professor Dunkerley for the paper we have just heard. It represents an enormous amount of work, and he has brought to light many new points, at any rate points new to a great many of us, and I am sure it is the wish of the meeting that we should offer our very best thanks to Professor Dunkerley for his kindness. I would ask him, if it is not encroaching too much on his time, to give us another year a paper investigating a case where the revolutions are very high; instead of being 125, they might be 400, as they are in a destroyer, or a torpedo boat, when the inertia stresses become very great. If he would do that, we should feel under a further debt of obligation to him, and it would make his investigations more complete and, therefore, all the more valuable.

* Mr. Macfarlane Gray regrets that pressure of other matters prevents his giving the above solution at present.—ED.

TORSIONAL VIBRATIONS OF SHAFTS.

By Herr L. Gümbel, Associate.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 20, 1902; A. F. Yarrow, Esq., Vice-President, in the Chair.]

The resolution, by the aid of Fourier's Theorem, of periodic into harmonic forces has already enabled us to solve a number of very important technical problems, such as the balancing of marine engines, the researches concerning the way in which the tangential forces of the engine can be rendered uniform, the problem of the pitching and rolling motion of ships, &c.

The paper I have the honour to lay before you to-day will demonstrate again how Fourier's Theorem helps us to open up a field which, as yet, has scarcely been accessible, namely, the study of the torsional vibrations of shafts.

Historical.

So far as I am aware, Dr. Bauer, of Stettin,* was the first to point out that there are torsional vibrations in the shafts of marine engines owing to the periodic fluctuations, both in the tangential forces acting on the cranks and in the propeller resistances. Dr. Bauer was led to this conclusion by actually measuring the angular velocities of shafts.

Professor Lorenz, of Göttingen,† has dealt with the matter mathematically. He does not take the propeller resistance into consideration, however, and hence does not deal with the actual conditions of the problem.

To Frahm,‡ of Hamburg, we owe a very extensive contribution to this problem, by experiments conducted with admirable care and great ingenuity. We may look forward with particular interest to the detailed account of these investigations.

* "Jahrbuch der Schiffbautechnischen Gesellschaft." Vol. I., 1900.

† "Dynamik der Kurbelgetriebe." Leipzig, 1901.

‡ Verhandlungen der Gesellschaft deutscher Naturforscher und Aerzte. Hamburg, 1901.

In your country, I see from *Engineering*, January, 1902, that Messrs. Julius Frith and E. H. Lamb have, in a paper read before the Manchester Section of the Institution of Electrical Engineers, drawn attention to the fact that vibrations may arise in the shafts of dynamos: I do not know how far their work has extended.

SYSTEM.

In order to attack the problem, we may imagine the system comprising engine, shaft, and propeller to be reduced to the following elements:—

(1) A mass m acting on the crank radius r at the middle of the crank shaft.

(2) A mass M acting on the crank radius r at the middle of the propeller boss.

The masses m and M have to be such that $m\,r^2$ and $M\,r^2$ represent the moments of inertia of the rotating masses of the engine and of the propeller respectively.

(3) The two masses m and M are supposed to be connected by a shaft without mass, of uniform diameter and of length L, measured from the middle of the crank shaft to the middle of the propeller boss. The shaft being assumed to be without mass, the angle of distortion of the shaft will be the same for all points of the shaft.

COMPOSITION OF MASSES.

The mass m will be composed of—

(a) One-third of the mass of the shaft, reduced to the crank radius, and reckoned from the centre of mass of the system up to the end of the shaft on the crank side.

PROOF.—Let ξ be the angle of torsion of a shaft section, at distance x from the centre of gravity. Since the centre of gravity remains at rest, when the shaft is in natural vibration, ξ will increase proportionately with x, so that $\xi = C\,x$. Let m_1 L be the mass of the shaft, and χ_m the angle measuring the distortion of the shaft at the crank end due to the mass acceleration of the shaft. Then—

$$d\,\chi = C_1\,m_1\,d\,x\,\omega^2\,r\,\xi\,x$$

$$\chi_m = C_1\,C\,m_1\,\omega^2\,r\int_0^{L_m} x^2\,d\,x$$

(where L_m stands for $\dfrac{L\,.\,M}{M + m}$)

$$= \frac{C_1 C}{3}\,m_1\,\omega^2\,r\,L_m^{\,3}.$$

The same value would result if we had only one mass $m_2 L_m$ on the shaft's end defined by the equations—

$$C_1 . m_2 . L_m . \omega^2 r . C . L_m . L_m = \frac{C_1 . C}{3} . m_1 \omega^2 r L_m^3.$$

$$m_2 L_m = \frac{m_1}{3} . L_m.$$

This substitution is strictly correct, of course, only for the natural vibrations of the shaft. The above term $\frac{L . M}{M + m} = L_m$ is the distance of the centre of gravity from the assumed end of the shaft on the crank side, the corresponding distance from the boss being $\frac{L . m}{M + m}$. (Fig. 1, Plate XVI.)

(*b*) The mass of the crank and crank pin reduced to the crank radius.

(*c*) The rotating part of the mass of the connecting rod $= \frac{m_c e_1 e_2}{l^2}$, where m_c is the mass of the connecting rod, l its length, e_1 the distance of the mass centre of the rod from the crosshead, e_2 the distance between the centre of the crosshead pin and the centre of the rod.

(*d*) A certain part of the reciprocating masses—

$$= \frac{2}{\pi} \cdot \left(m_c \cdot \frac{(l - e_1)}{l} + m_p\right)$$

where m_p is the sum of the masses of crosshead, piston rod, and piston.

The mass M is composed of—

(*a*) One-third of the mass of the shaft reduced to the crank radius and measured from the centre of gravity of the system to the propeller end of the shaft.

(*b*) The mass of the propeller and the boss, reduced to the crank radius.

Forces Acting on the System.

On the system as sketched out, the following forces are acting:—

(1) On the crank side acting on the crank radius—

(*a*) The tangential forces resulting from the steam pressure on the piston.

(*b*) The accelerating forces of the reciprocating parts, under the supposition of a constant mean angular velocity.

The tangential force (*a*) due to the steam pressure is composed of a constant mean force and of periodic forces. The mass acceleration components (*b*) being likewise periodic variables, we have means at hand to equalise the non-uniformity of the tangential steam force by the mass acceleration, and this method of restoring uniformity is indeed in general practical use. As Professor Lorenz * has, however, asserted that the method could not be recommended, because the balance would be disturbed by any alteration of the engine speed, I should like to point out here that in marine engines the tangential forces of the steam pressure follow practically the same law as the mass accelerations. Both are, with close approximation, proportional to the square of the number of revolutions. A change in the speed will, therefore, not affect the balance between the two forces.

(2) On the propeller side acting on the crank radius—

(*a*) A constant mean resistance corresponding to a constant angular velocity of the propeller.

(*b*) Periodically variable forces of resistance corresponding to the variable angular velocity of the propeller.

(3) There will be distributed over the whole length of the shaft damping forces, due to the internal friction in the shaft under torsion, and proportional to the angle of distortion and to the velocity of distortion. These forces can be resolved into a couple of equal and opposite forces, acting on the crank radius at the two ends of the shaft.

(4) Further, there are acting at the two ends of the shaft the accelerating forces of the revolving masses m and M, whose intensity will depend upon

(*a*) The angle α, or β, respectively, of the end sections.

(*b*) The frequency of the vibrations.

The amplitudes of the forces will be $m\,\omega^2\,r\,\alpha$ and $M\,\omega^2\,r\,\beta$ respectively.

Fig. 3 (Plate XVI.) represents this system of forces for a definite period.

Law of the Propeller Resistance.

If we disregard any special external influences, and assume the propeller to be revolving in open water, the variable resistance opposed to the propeller will depend exclusively upon its angular velocity.

Fig. 2 (Plate XVI.) is based upon Rota's experiments † concerning the magnitude of the propeller thrust T for constant ship speeds and variable propeller revolutions. Rota's law may be expressed by the equation—

$$T = -T_m + C\,(v_x - 0{\cdot}3\,v_m)^2.$$

The propeller speed v_m corresponds to the condition of equilibrium. For this value v_m, the propeller thrust is equal to the ship's resistance: $T_m = C\,.\,0{\cdot}7^2\,v_m{}^2$. When

* Transactions of the Institution of Naval Architects, 1900.

† Rota: "La Vasca per l'Esperienze di Architettura navale." Genova, 1898.

the propeller speed increases (or decreases), the ship's speed remaining constant—which is approximately our case—the propeller thrust will increase (or diminish) in accordance with the equation—

$$\Delta T = \frac{2 T_m \Delta v_m}{0{\cdot}7 v_m}.$$

As long as the increase or decrease in v_m does not constitute any considerable part of v_m—as is the case in our problem—the increase in the thrust may be regarded as directly proportional to the increase in the propeller speed. Assuming a proportionality between the propeller thrust and the turning moment of the propeller, we may likewise consider the amplitudes of the periodic propeller resistance as proportional to the amplitudes of the periodic fluctuations in the propeller speed.

If, therefore, $\omega r \beta$ be the amplitude of the propeller speed, the amplitude of the propeller resistance will be $k \omega r \beta$.

Analysis.

Whatever may be the law governing the tangential force of the engine, we can always, by the aid of Fourier's Theorem, resolve this force into a constant mean force P, and into harmonic forces whose frequency will be a multiple of the number of the revolutions made by the engine, thus—

$$P = P_m + P_1 \cos (c_1 + x) + P_2 \cos 2 (c_2 + x) + P_3 \cos 3 (c_3 + x) + P_4 \cos 4 (c_4 + x) \ldots,$$

where P_1, P_2, P_3 . . . represent the amplitudes of the harmonic forces, and the angles c_1, c_2, c_3 . . . the phases of the harmonic forces at the moment $x = 0$.

As regards the determination of the values of the P and c, I would refer you to Perry's excellent "Calculus for Engineers."

The mean force P_m being constant, there will be a constant angular velocity of the propeller, and a constant angle of torsion of the shaft. On the other hand, periodically variable propeller speeds and distorsions of the shaft will result from the periodic forces.

The forces R which are due to this varying angular velocity of the propeller may likewise be expressed in a Fourier series of the form—

$$R = R_m + R_1 \cos (k_1 + x) + R_2 \cos 2 (k_2 + x) + R_3 \cos 3 (k_3 + x) + R_4 \cos 4 (k_4 + x) + \ldots$$

In this expression may also be included forces occasioned by external circumstances such as shocks, the passing of the hull of another ship near the propeller, &c., provided they recur at regular periods.

Conditions of Equilibrium.

The conditions of equilibrium between the forces of the engine and of the propeller, demand that forces of equal frequency should be equal to one another, namely—

$$P_m = R_m$$
$$P_1 \cos (c_1 + x) = R_1 \cos (k_1 + x),$$
$$P_2 \cos 2 (c_2 + x) = R_2 \cos 2 (k_2 + x), \text{ \&c.}$$

Solution of the Problem.

To maintain equilibrium in the system—as indicated above—the geometric resultant of all the forces of the same frequency must become zero, since all the forces vary harmonically. This condition cannot be fulfilled without our knowing the phases of the component forces for any particular moment.

Let us once more, by the aid of Fig. 4 (Plate XVI.), review the component forces from this point of view. At the propeller end of the shaft there are acting on the crank radius r—

(1) The accelerating force $M \omega^2 r \beta$;

(2) The propeller resistance $k \omega r \beta$, which will attain its maximum at the moment of maximum speed; this vibration lags, therefore, by 90° behind the variable $M \omega^2 r \beta$;

(3) The damping force $k_1 \omega r \dfrac{a \sin \phi + \beta}{\sin \epsilon}$.

The maximum of this component coincides with the maximum speed of deformation, and it will therefore lag by 90° behind the deformation itself.

At the engine end of the shaft there are acting on the crank radius r—

(1) The accelerating force $m \omega^2 r a$;

(2) The tangential force P.

(3) The damping force $k_1 \omega r \dfrac{a \sin \phi + \beta}{\sin \epsilon}$. This component will be equal to the corresponding force acting at the propeller end of the shaft, but of opposite direction. The forces acting at the propeller end must at any moment be in equilibrium with the internal torsional forces in the shaft. The same holds good for the forces in play at the engine end of the shaft. It follows, then, that the resultant of the forces at the one end must be equal in magnitude, but of opposite sign to the resultant at the other end. We thus arrive at the force polygon, of which Fig. 4 (Plate XVI.) gives a diagram; it explains the relations existing between the different components.

Composition of the Vibration out of Three Partial Vibrations.

The vibrations of the terminals α and β are not pure torsional vibrations, but are composed of three partial vibrations.

(*a*) A torsional vibration, whose amplitude is indicated by the angle η or ζ respectively.

(*b*) A vibration of amplitude γ, which is the same for all the points of the shaft.

(*c*) A vibration, at 90° in advance of the torsional vibration, of amplitude λ or ν respectively.

The sub-division of the vibrations α and β into these components is arbitrary, and the magnitude of the components can be varied by changing the phases which the angles η and γ, and ζ and γ respectively interfere.

The sum of the torsional angles $\eta + \zeta$ is, of course, not variable.

Two ways of dividing α and β deserve particular attention.

(*a*) There is a phase difference of 90° between η and γ, and ζ and γ respectively. In this case, the resolution will yield the minimum value for the vibration γ in which all the points of the shaft partake equally.

(*b*) This vibration γ, which is common to all the points of the shaft, is supposed to be 90° in advance of the propeller vibration β. This is the assumption adopted in our calculations, for reasons presently to be explained. I wish to emphasise once more that any other assumption might be made, and that the values of α and β, and of $\eta + \zeta$ are not influenced by the particular selection made. From Fig. 4 (Plate XVI.) the following equations may at once be deduced:—

$$(1)\quad \gamma = \frac{k}{M \cdot \omega} \cdot \beta.$$

$$(2)\quad \cot \varepsilon = \frac{k}{M\,\omega}.$$

$$(3)\quad \tan \phi = \frac{1}{\cot \varepsilon} \cdot \left(1 - \frac{C}{\sin \varepsilon \cdot \omega \cdot \sqrt{k^2 + M^2\,\omega^2}}\right).$$

$$(4)\quad \beta\,\omega\,\sqrt{k^2 + M^2\,\omega^2} \cdot \sin \varepsilon = C\,\omega\,(\alpha \sin \phi + \beta).$$

$$(5)\quad \alpha = \gamma \cdot \frac{\sin \varepsilon}{\sin (\varepsilon - \phi)}.$$

$$(6)\quad \sin \delta = \frac{k\,\omega\,r\,\beta - m\,\omega^2\,r\,\alpha \cdot \cos \phi}{P}.$$

$$(7)\quad \frac{P \cos \delta + m\,\omega^2\,r\,\alpha \sin \phi}{M\,\omega^2\,r} = \beta.$$

The co-efficient C expresses the ratio of the torsional force acting upon the shaft to the angle of torsion—

$$(8) \qquad C = \frac{I \,.\, G}{L \,.\, r^2},$$

where I is the polar moment of inertia of a shaft section, G the modulus of elasticity, and L the length of the shaft.

From the equations (1) to (8) it follows that—

$$(9) \qquad \eta = \frac{a \,.\, \sin \phi}{\sin \epsilon} \,.\, \cos \mu.$$

$$(10) \qquad \zeta = \frac{\beta}{\sin \epsilon} \,.\, \cos \mu.$$

$$(11) \qquad \text{where } \mu = \frac{k_1 \,.\, \omega}{C}.$$

For any given value of ω, the equations (1) to (11) can be solved for a given system of masses and forces.

Vibration of any Point in the Shaft.

The amplitude and phase of the vibration of a point in the shaft at the distance x from the mass m will be found as the resultant of the following vibrations:—

(1) A vibration of the amplitude $\frac{L \,.\, \eta - x(\eta + \zeta)}{L}$, of the same or opposite phase of η.

(2) A vibration of the amplitude and phase of γ.

(3) A vibration of the amplitude $\frac{L \,.\, \lambda - x(\lambda + \nu)}{L}$, of the same or opposite phase of λ.

Example.

The internal damping forces have not been taken into consideration in the following calculations, as the value k_1 seems to be rather small in comparison to the value k of the propeller resistance.

Hence $\cos \mu = 1$, and Fig. 4 alters to Fig. 5 (Plate XVI.).

By the aid of formulæ (1) to (10), we can calculate all the vibration values for a given force of any periodicity, having given the masses m and M, and the shaft dimensions, when the propeller resistance k is known. Figs. 8, 9, 10 (Plates XVII. and XVIII.) illustrate the results of such calculations for the following data:—

$$m = \frac{10670}{9 \cdot 81} \; \frac{k\,g}{m} \sec^2.$$

$$M = \frac{27860}{9 \cdot 81} \; \frac{k\,g}{m} \sec^2.$$

$$r = 61\,cm.$$

$$L = 5248\,cm.$$

$$I = 103000\,cm^4.$$

$$G = 850000 \frac{k\,g}{c\,m^2}.$$

Fig. 9 (Plate XVIII.) gives the values of the co-efficient of the propeller resistance k and Fig. 8 (Plate XVII.) the periodic force P for frequencies of from 150 to 290 periods per minute, the frequency of P having been assumed to be four times the number of revolutions of the engine.

Fig. 8 reproduces the force polygons for different frequencies; the same figure marks the forces $M\,\omega^2\,r\,\beta$, $m\,\omega^2\,r\,\alpha$, and $k\,\omega\,r\,\beta$. Both the intensities and the phase differences of these forces change with the number of periods. Force and torsional vibration will be in the same phase for the speed 0 and forces 0; they will be in opposite phase, when the speed becomes $= \infty$.

SPECIAL CASE $\delta = -90°$.

The frequency for which the angle δ becomes $-90°$ is of especial interest. In this case we have (compare Fig. 6, Plate XVI.)—

$$\frac{m\,\omega^2\,r\,\gamma}{M\,\omega^2\,r\,\gamma} = \frac{m\,\omega^2\,r\,\beta \sin \epsilon}{\sin \epsilon \,.\, C\,r\,(\alpha \sin \phi + \beta)},$$

and, therefore, as—

$$m\,\omega^2\,r\,\alpha \sin \phi = M\,\omega^2\,r\,\beta.$$

$$C\,r\,(m + M) = m \,.\, M \,.\, \omega^2\,r,$$

$$\omega = \sqrt{\frac{C\,(M + m)}{M\,m}},$$

$$C = \frac{I \,.\, G}{L \,.\, r^2},$$

$$\omega = \sqrt{\frac{I \,.\, G \,.\, (M + m)}{L\,r^2\,M\,m}},$$

NATURAL VIBRATIONS.

If we imagine the whole system to be in free natural vibration, without any opposing resistance, the centre of gravity of the whole system must remain unaffected by the vibrations of the component parts. Hence we may assume the shaft to be clamped at its centre of gravity. We then have two systems, namely, a mass m vibrating at one end at a distance $\frac{L \cdot M}{M + m}$ from the centre of gravity, and a mass M at the other end at a distance $\frac{L \cdot m}{M + m}$ from the centre of gravity.

Now $m\,\omega^2\,r\,\alpha = C_\alpha\,r\,\alpha$, and $M\,\omega^2\,r\,\beta = C_\beta\,r\,\beta$, and, further—

$$C_\alpha = \frac{I \cdot G}{L \cdot \frac{M}{M+m} \cdot r^2} \quad \text{and} \quad C_\beta = \frac{I \cdot G}{L \cdot \frac{m}{M+m} \cdot r^2}$$

hence, in both cases—

$$\omega = \sqrt{\frac{I \cdot G \cdot (M+m)}{L\,r^2\,M\,m}}.$$

This ω represents the angular velocity of the natural vibration of the system.

We see that this value of ω, corresponding to the natural vibration, is the same as that for which $\delta = -90°$.

IMPORTANT FORMULÆ.

The vibration plotted in Fig. 6 (Plate XVI.) is therefore composed of the following partial vibrations:—

(1) A vibration of amplitude γ, in which all the elements of the shaft participate; the accelerating forces of this component are, at any moment, in equilibrium both with the damping force $k\,\omega\,r\,\beta$, and with the force P.

(2) A vibration exactly like the free vibration of the shaft, unimpeded by any resistance. Whilst, however, the amplitude of a free vibration is indefinite, in the absence of any resistance, the damped vibrations in our case have a definite amplitude, which, as Fig. 6 indicates, will be, for the propeller end—

$$\beta = \frac{P\,M}{m\,k\,\omega\,r},$$

and for the engine end of the shaft—

$$\alpha = \frac{\sqrt{k^2 + M^2\,\omega^2}}{m\,\omega} \cdot \beta = \frac{P \cdot M \cdot \sqrt{k^2 + M^2\,\omega^2}}{m^2\,\omega^2\,r\,k}.$$

The angle of torsion of the two end sections of the shaft will be—

$$\eta + \zeta = \frac{P \cdot (M+m)\,\sqrt{k^2 + M^2\,\omega^2}}{k\,\omega^2\,r\,m^2}.$$

These formulæ are of very great practical importance.

The resolution of the vibrations α and β, which we performed above, into torsional vibrations η and ζ, and a vibration γ of all the shaft elements at right angles to β, has been made with special regard to this case.

We recognise in Fig. 10 (Plate XVIII.) that the maximum amplitude of vibration does not quite coincide with the natural period of vibration of the shaft. This coincidence, in fact, occurs already at a slightly lower number of vibrations. In practice, however, the above formulæ will be found sufficient for the calculation of the critical values of α, β, and $(\eta + \zeta)$.

Forced Vibrations without Damping.

Those general formulæ further enable us to deduce special formulæ for the vibrations which, in the absence of damping, would result under the influence of the harmonic force P. In that case k will be 0.

We see from equations (2) and (3)—

$$\varepsilon = \phi = 0°.$$

and from equation (4)—

$$\beta . M \omega^2 r = C . r (\alpha + \beta),$$

and from equation (7)—

$$\beta . M \omega^2 r = P + m \omega^2 r \alpha.$$

and that : (1) the sum of all the external forces is zero ; (2) the external forces are in equilibrum with the internal forces. Also $M \omega^2 r \beta$ and $m \omega^2 r \alpha$ vibrate in opposite phases.

As long as the number of periods is less than the natural frequency, P will vibrate in phase with $m \omega^2 r \alpha$; when the natural frequency is exceeded, P will pass over to the contrary phase.

We further deduce from those two equations—

$$\beta = \zeta = \frac{P . C}{(M + m) . C . \omega^2 r - M m \omega^4 r}.$$

$$\alpha = \eta = \frac{P (M \omega^2 - C)}{(M + m) . C . \omega^2 r - M m \omega^4 r}.$$

The values of α and β have been calculated for the masses stated above, and the results have been marked on Fig. 10 (Plate XVIII.).

Both β and α become $= \infty$, when—

$$(M + m) . C \omega^2 - M m \omega^4 = 0,$$

that is, when—

$$\omega = \sqrt{\frac{C(M + m)}{M m}},$$

i.e., for the number of periods corresponding to the natural vibration of the shaft.

The angle δ alters suddenly from 0° to 180° by passing the frequency of natural vibrations (Fig. 9, Plate XVIII.).

Vibration of the Centre of Mass.

Some further interesting conclusions concerning the vibration of the centre of mass of the system may be derived from the polygon of forces of Fig. 5 (Plate XVI.). The vibration of the centre of gravity is the result of the composition of the torsional vibration $\frac{M \zeta - m \eta}{M + m}$, and of the vibration common to all the points of the shaft, γ.

If we trace, in the polygon of forces (Fig. 5), the respective acceleration forces $(M \omega^2 r \zeta - m \omega^2 r \eta)$ and $(M + m) \omega^2 r \gamma$, we recognise that the resultant of these forces is equal and opposite to the resultant of the external forces P and $k \omega r \beta$, and we further find the rule confirmed, that—whatever the internal forces of a system—the centre of mass will always move as if all the external forces distributed over the system were acting at that centre.

Solution of an Example when a Periodic Force is Given.

Having shown how the amplitudes of the vibrations can be calculated for a given force of any periodicity, we will now follow the course of the amplitudes during one period for a definite speed, the shaft being acted upon by any given periodic force.

Fig. 11 (Plate XIX.) shows the resulting tangential force diagram of an engine, which is developed in the Fourier series—

$$\begin{aligned} P \text{ (in kg.)} = 34760 &+ 1400 \cos (240 + x) + 7000 \cos 2 (10 + x) \\ &+ 3400 \cos 3 (20 + x) + 2410 \cos 4 x. \end{aligned}$$

The engine speed is $n = 70$ revolutions per minute. Assuming that 90 per cent. only of the mean force, 34,760 kg., are taken up by the propeller, the co-efficient of the propeller resistance will be—

$$k = \frac{2 \cdot 0{\cdot}9 \cdot 34760 \cdot 30}{0{\cdot}7 \cdot 70 \cdot 3{\cdot}14 \cdot 61} = 200.$$

We will further suppose that the periodic forces are reduced by 5 per cent. before they enter into action at the crank radius, and that they are thus expressed by the series—

$$33000 + 1330 \cos (240 + x) + 6650 \cos 2 (10 + x) + 3230 \cos 3 (20 + x) + 2290 \cos 4 x.$$

The amplitudes of the vibrations can be determined in the manner already explained, and we thus obtain the following Table :—

Number of Periods.	Simple.	Twofold.	Threefold.	Fourfold.
P, in kg.	1,330	6,650	3,230	2,290
β	0·00946	0·01993	0·01550	0·00204
α	0·00696	0·0149	0·03375	0·00949
$\eta + \zeta$	0·00445	0·0301	0·0460	0·01145
ϵ	46° 12′	64° 23′	72° 43′	76° 32′
ϕ	−63° 41′	29° 2′	64° 40′	73° 40′
δ	33° 40′	14° 53′	−8° 41′	−159° 42′

The polygons of the forces and of the amplitudes $\eta + \xi$, α, β, are reproduced in Fig. 7 (Plate XVI.).

When we plot the vibrations of the separate periods in a rectangular system of co-ordinates, we find the resulting vibration by algebraical computation of the component amplitudes of that moment. Since we know the phase difference between each partial vibration and the respective force, we can also express the periodic vibration laws algebraically in a Fourier series.

We find—

$$\beta = 0{\cdot}00946 \cos(93^\circ\,40' + x) + 0{\cdot}01993 \cos 2\,(107^\circ\,26' + x) + 0{\cdot}0155 \cos 3\,(77^\circ\,6' + x) + 0{\cdot}002035 \cos 4\,(5^\circ\,4' + x).$$

$$\alpha = 0{\cdot}121 + 0{\cdot}00696 \cos(120^\circ + x) + 0{\cdot}0149 \cos 2\,(-13^\circ\,2' + x) + 0{\cdot}03375 \cos 3\,(8^\circ\,40' + x) + 0{\cdot}00949 \cos 4\,(-41^\circ + x).$$

$$\eta + \zeta = 0{\cdot}121 + 0{\cdot}00445 \cos(229^\circ\,52' + x) + 0{\cdot}0301 \cos 2\,(4^\circ\,38' + x) + 0{\cdot}0460 \cos 3\,(11^\circ\,20' + x) + 0{\cdot}01145 \cos 4\,(-43^\circ\,18' + x).$$

In these formulæ, 0·121 represents the constant angle of torsion between the two end sections, due to the constant tangential force of 33,000 kg. $= 0{\cdot}95 \times 34{,}760$ kg.

The curves for α, β, and $\eta + \zeta$, of Figs. 12, 13, 14 (Plate XIX.) should, I think, demonstrate with sufficient clearness, how much the law of the distortion of the shaft, when derived dynamically, deviates from the conclusions to which statical considerations lead us even under conditions like those of our example, when we have to deal with forced vibrations, somewhat remote, as regards their frequency, from the natural vibrations of the shaft.

Before finishing this paper, I may be allowed to point out that the same method which is here applied to problems concerning marine engines may be used in the case of any other engine, especially for dynamos, for which we may likewise assume a proportionality between the turning moment and the angular velocity.

Conclusion.

Two rules may be laid down as self-evident conclusions to be drawn from these deductions, already stated by the above-mentioned authors.

(1) The force with which the engine acts on the shaft should be kept as uniform as possible.

(2) We should avoid the continued application of revolutions of the engine whose number bears a simple ratio to the frequency of the natural vibration of the shaft.

The following addition may, in my opinion, be made to these simple rules :—

(3) We should determine the possible tangential force diagrams for that number of revolutions, likely to be adopted in practice, which is most exposed to vibrations, and then ascertain, with the aid of the formulæ given, whether the resulting distortions $(\eta + \zeta)$ of the shaft will remain within admissible limits.

DISCUSSION.

Mr. W. E. Smith (Member of Council): Mr. Chairman and Gentlemen, I am, as you all know, not a technical engineer, but I am engaged in installing engines on board our ships, and am, consequently, much interested in their behaviour. We all remember that when we went to school, years ago, we were taught what is called the third law of motion, viz., that to every action there is a corresponding opposite and equal reaction. This study of the torsional vibration of shafts has an important bearing on the torsional vibration of ships caused thereby. We all know that the torsional vibration on ships is not nearly so large in amount as the longitudinal vibrations caused by the forces which have just been dealt with at some length by Professor Dunkerley. It was not so very long ago in point of years, although a considerable time ago in point of march of events, that we were driven, as naval architects, to resort to all sorts of impromptu appliances for reducing the longitudinal vibrations of ships. I remember, in the old *Rattlesnake* days, dealing with considerable vibration of the deck over the engine-room. That deck was supported, as is usual, by beams of uniform sectional area, and, as the engines could not be altered, we had to alter the deck. We resorted to the artifice of making one beam twice the depth of the next, and that next beam one and three-quarters the depth of the next, and so on, purposely throwing the beams out of tune, and making it difficult for them to vibrate in the same time. The plan succeeded, but the cause remained, although the effect was removed. In those days, engineers were not very keen to take up this question about the reduction of the causes of vibrations of ships; but, one after another, they all studied the question, and ultimately, as regards the longitudinal vibration of the ships, arrived at what we now call the Yarrow-Schlick-Tweedy system, or some similar system, and practically the whole longitudinal vibration is cured. We have, for instance, just passed into the Navy a ship of almost unprecedented power. She has most successfully gone through her trials as regards longitudinal vibration, and on account of the great care bestowed in fixing the angles of the different cranks, in determining their powers, in suitably disposing the several cylinders, and all that sort of thing, we have got practically complete absence of longitudinal vibration. Up to the present, the question of torsional vibration has not received the same amount of extended investigation, and that is quite natural, on account of what I said before, viz., that it is much less in amount; but still, it has an important bearing with regard to warships, although, perhaps, not more important in warships than it may possibly be in some other special classes of ships. We all hear a great deal nowadays about the man behind the gun, and how much we have to trust to him. I assure you, from having seen the thing myself, that this question of torsional vibration on ships is a matter that, under certain circumstances, may affect the accurate shooting of guns when trained on the beam. When a ship is rolling at sea, the broader oscillations can be allowed for, and the gunner very soon acquires the skill for that. and for letting the gun off at the proper time; but with a ship that has, under certain conditions, a very pronounced torsional vibration, the motion is so quick in period, that it is extremely difficult for a man, under the circumstances, to allow for. This introduces a difficulty in shooting which we, as naval architects, think ought not to exist, and we should be very grateful for any investigation which would tend to reduce it. As to the influence that this question of reduction of torsional vibration may have on the saving of weight, we cannot hope for very much saving, because the forces are not very large in amount. A small amount of structural weight put in the proper place may, perhaps, very largely damp the oscillation, but we must not think too lightly of the problem simply because the forces are small. We should think them out and get rid of the trouble. The principle laid down on the last page of the paper is one that, I think, all engineers must heartily concur in. Herr Gümbel says, speaking of torsional vibrations of ships (although this applies generally), "that the force with which the engine acts on the shaft should be kept as uniform as possible." That is in the direct interest of saving of weight. The next statement is also one which we must cordially endorse, and it is very clearly put, viz., that "we should avoid the continued application of revolutions of the engines which bear a simple ratio to the frequency of the natural vibration of the ship." In this paper, that has reference only to the torsional vibrations, although it applies really to other things as well. I hope some technical engineer will come forward and discuss the problem purely from an engineering point of view, but, as a naval architect, I feel so much gratitude to our mathematical engineers who take the trouble to investigate these problems, and help us out of the difficulties which we should otherwise be in, that I could not refrain from rising to thank Herr Gümbel for the great pains he has taken, and to assure him of the undoubted benefits which all designers of ships, who take a proper interest in their business, derive from such investigations as these.

Mr. Macfarlane Gray (Member of Council): Herr Gümbel has paid our Institution a high compliment in presenting this paper to us. Not many of us will master all its details, but Mr.

Smith seems to have a clear grasp of the paper, and speaks highly of it. I can only admire the mathematical talent which the author has disclosed in these long investigations, though I cannot pretend to have followed them in detail.

Professor S. DUNKERLEY (Associate): I am afraid, Sir, that I cannot profitably join in the discussion, because time has only permitted me to glance hurriedly through Herr Gümbel's paper. But I certainly think that it is one of great value, and I should be sorry to miss this opportunity of expressing my appreciation of it. The subject of torsional vibrations of shafts, subjected to periodically varying turning moment and resistance, is exceedingly complicated, and the present paper is, I think, a very long step towards the correct solution of the problem.

Herr L. GÜMBEL (Associate): I beg to thank you, Gentlemen, for the kind reception you have afforded me, and I hope that the small theoretical contribution I have had the honour to lay before you may conduce to further the solution of a highly practical engineering problem—that of the safety of shafts.

The CHAIRMAN (Mr. A. F. Yarrow, Vice-President): On behalf of the Institution, I have much pleasure in thanking Herr Gümbel for his very interesting paper, as it is one of considerable value.

IMPROVEMENTS IN PROPELLER SHAFT BEARINGS.

By A. Scott Younger, Esq., B.Sc., Member.

[Read at the Spring Meetings of the Forty-Third Session of the Institution of Naval Architects, March 20, 1902; A. F. Yarrow, Esq., Vice-President, in the Chair.]

At the Spring Meetings of this Institution in 1900, the writer had the honour of reading a paper on "Failure and Corrosion of Propeller Shafts," in which an attempt was made to show that many of the recent failures of the propeller shafts of cargo steamers were partly to be explained by the bending of the shaft due to the weight and thrust of the propeller, taken in conjunction with the severe stresses induced by long voyages in ballast. It was shown that this bending action produced its greatest effect at the ends of the brass liners, just where most of the fractures occur; and one of the conclusions arrived at was that the liners on the shaft were a source of weakness in producing a sudden break in the strength of the shaft, and thus localising the action at the change of section.

By way of illustrating this action, the attention of the meeting is directed to the Table on the next page, giving the results obtained from fourteen steamers, of which a careful record has been kept. All these vessels have been built within the last seven years, and are of the regular cargo type, having speeds from 8 to 10 knots. They have been employed in all trades, and have made many voyages to all parts of the world, loaded and light. This Table is extremely interesting and instructive, and certainly appears to justify the remark that some alteration in existing practice with regard to propeller shafts is necessary.

These shafts were all made of iron by good makers, and, with three exceptions, were condemned on account of the circular fracture which frequently appears at the ends of the after liner. Of these three exceptions F is still at work, and when last examined, in August, 1901, was in fairly good order, but showed distinct evidences of the well-known grooving action. The sister ship E met with a mishap to her propeller, or, doubtless, her shaft would have been still at work. The third exception is H, which broke at sea, involving the towage of a fully laden ship some 800 miles. Curiously enough, this shaft was examined about the same time as that of her sister ship

I; but, being apparently in good order, was passed by Lloyd's surveyor. It will thus be noticed that this shaft broke within two years of its being examined and passed. All the other shafts were condemned in dry dock.

RECORD OF 14 PROPELLER SHAFTS.

Steamers.	Dead Weight.	Water Ballast.	Weight of Propeller	Diameter of Shaft as fitted.*	Life of Shaft, in years.	Life of Shaft, in Miles Steamed.			Remarks.
						Loaded.	Ballast.†	Total.	
	Tons.	Tons.	Tons.	In.					
A	7,000	1,213	12½	13¾	2¾	52,800	57,300	110,100	Three sister ships; A and B being built by one firm. Mean light draughts, 13 ft. to 15 ft.
B	7,000	1,213	12½	13¾	2⅝	59,300	28,900	88,200	
C	7,000	1,228	11	13⅜	2⅛	37,700	48,700	86,400	
D	6,400	956	8½	13¼	1¼	21,900	35,000	56,900	Mean light draught, 12 ft. to 14 ft. 6 in.
E	6,200	1,172	9½	13	4½	123,100	68,000	191,100	Three sister ships; E and F being built by the same firm. E lost three propeller blades, which caused severe damage to stern tube, and necessitated new shaft. G has 240 tons less water ballast and ½ in. smaller shaft than E and F.
F	6,200	1,172	9½	13	5 years old, still at work	144,000	47,000	At 8/01, 191,000	
G	6,200	932	8½	12½	3⅛	84,300	46,300	130,600	
H	5,500	904	8¼	12	4¾	125,000	53,900	178,900	Two sister ships, built by same firm. Mean light draught, 12 ft. to 14 ft.
I	5,500	904	8¼	12	3	78,000	31,400	109,400	
J	5,300	870	8	12	4	131,000	29,600	160,600	Two sister ships, built by same firm. Mean light draught, 12 ft. to 14 ft.
K	5,300	870	8	12	3¾	103,200	53,100	156,300	
L	4,570	744	7½	12¼	3¼	78,100	51,800	129,900	Mean light draught, 11 ft. to 12 ft.
M	4,360	780	7½	11½	2¼	76,600	17,200	93,800	Mean light draught, 11 ft. to 12 ft.
N	3,960	680	7½	12	4⅛	83,600	28,100	111,700	Mean light draught, about 12 ft.

* In all cases, these sizes are in excess of Lloyd's requirements at the date of building.
† In addition to their water ballast, the vessels had, in many cases, bunker coals for a long voyage.

The first group, A, B, C, are wonderfully uniform, and call for little remark, beyond a reference to the low average life of the shaft.

The next group, E, F, G, illustrate the advantage of extra ballast and increased diameter of shaft. Here it will be noticed that G has 240 tons less water ballast and ½ in. less diameter of shaft than her sister ships, and it seems fair to conclude that the short life of her shaft was partly due to the combination of these qualities. The same remark applies to the case of D. The remaining cases appear to call for no special comment, if we except the remarkable uniformity in the short life of the shafts.

In view of the facts contained in the above Table it is questioned whether propeller shafts should not be more frequently examined than is the case at present. Lloyd's

Register require an inspection every two years; but even this limit seems too great for the extremely full steamers of the present day. The firm for whom the writer acts as superintendent now insist on having an annual inspection, if at all practicable.

The reasons for these unsatisfactory results being now pretty well understood, it is not so common as it once was to hear the failures of propeller shafts referred to as mysterious. The main reason is that too much has been expected from the propeller shaft, and the conditions under which it has to perform its work have gradually been made more and more severe.

Thus, while the size and fulness of steamers have increased, the engines and propeller shafts have not increased in anything like the same ratio. Again, the requirements of trade have been such, that many light-draught cargo steamers, having large propellers with comparatively small shafts, have had to make long ballast voyages.

Though, perhaps, a little surprising that the effect of these changes should not have been appreciated sooner, it is satisfactory to note that the registration societies have recognised the necessity for larger shafts and more frequent inspection.

It is also generally admitted that it would be an immense advantage if more ballast were carried, as any additional immersion would tell in favour of the propeller shaft, and modify the severe stresses occasioned by ballast voyages.

In the discussion which followed the author's paper of 1900, attention was called to the usual design of having the propeller entirely overhung, and with the shaft working in a slack bearing with water lubrication. There is no doubt that this is a defective arrangement, and must be held accountable for some of the trouble recently experienced.

Among the proposals recommended on that occasion as likely to lead to improvement in the method of supporting propeller shafts, there were two involving alterations and additions to the present arrangement, and of these, the one of adding a bearing aft of the propeller seemed, at first sight, the more promising, and, in some respects, the more scientific.

However, on further consideration it appeared that this plan could not be carried out without elaborate and costly alterations in the present design of the stern frame, though something may be done to reduce the overhang, by extending the bearing close up to, or even partially into, the propeller boss, and a design embodying this idea is referred to later on.

The other method is to abolish the brass liners, and run the shaft in a metal bearing lubricated with oil, and from which the water is excluded. The necessity for this last requirement becomes apparent when it is remembered that the vessel might remain two or three weeks in port without turning the engines, and if water had access to the shaft, considerable rusting would occur, and inevitably lead to trouble in the bearings when the voyage was resumed.

There have been many cases in which shafts have been run without liners ; but, so far as the author knows, there is no published record of the results obtained. As, however, several instances have recently come to his knowledge in which attempts have been made to run propeller shafts without liners and lubricated with oil, it seemed desirable that the results of some of these full-scale experiments should be brought together and given a permanent form.

The objects of this paper are :—

(1) To record some of these results;

(2) To bring before this Institution some improvements in propeller shaft bearings including :—

(*a*) A modified design of stern tube, with a stuffing-box at the after end, adjustable from within the vessel ; and

(*b*) A new design of after packing gland, which is considered suitable for linerless propeller shafts.

In designing these improvements, the writer has had associated with him his former colleague at the British Corporation Registry, Mr. J. Foster King, and the opportunity is here taken to acknowledge the valuable assistance received during the course of the work.

S.S. *Carlisle*.—The owner of this steamer, Mr. Simpson, of Cardiff, being interested in the propeller shaft problem, determined to try the effect of running the shaft without liners, in a stern tube filled with lubricant, but without having any gland at the after end. A new iron shaft was made equal in diameter to that of the old shaft over the liners, and fitted at Cardiff in March, 1900. A sketch of the arrangement is shown in Fig. 1 (Plate XX.). The lower two-thirds of the bearing was lined with white metal, the upper third having lignum vitæ staves fitted as usual, and the tube was filled before leaving dry dock with melted sternoline, or solidified oil. It will be noted that the only change from the usual arrangement consisted in having white metal in the bearing and oil in the tube. In the course of the voyage, which had 72 steaming days, 10 gallons of oil were put into the tube from a pipe in the saloon, being about one quart of oil every two days. At the end of the voyage, the steamer was dry-docked and the shaft was found to have worn down the bearing $\frac{1}{16}$ in. The tube was again run up with lubricant, and the engineer instructed to use double the quantity on the second voyage, which lasted about the same time as the first. Accordingly, the oil used amounted to about 20 gallons.

The third voyage also lasted about 4½ months, and, acting according to instructions, the engineer used about 35 gallons of oil, this being somewhat less than two quarts for every steaming day.

At the end of the third voyage, the steamer was again dry-docked, when the bearing was found to have worn down $\frac{3}{16}$ in. On this occasion the propeller was disconnected and the shaft drawn in for examination. It was found to be in good order throughout its length, except the last 12 in. at the after end of the bearing, which was somewhat pitted; also the part next the propeller, which was exposed to the action of the water, was corroded, and slightly reduced in diameter.

The results of this experiment may be summarised as follows:—

Duration of experiment	...	13 months.
Material of shaft ...	...	Iron.
„ „ bearing...	...	White metal.
Lubricant	...	Solidified oil with free access of water.
Wear down	...	$\frac{3}{16}$ in.
Condition of shaft ...	...	Good throughout the body, but defective at after end.

The shaft was again drawn in for examination in December, 1901, after travelling a further 27,000 miles, but little difference could be detected in its condition.

The conclusion to be drawn from this appears to be that, in order to make this method a success, it is advisable to exclude the salt water from the bearing, and also from the part of the shaft between the bearing and the propeller.

Some extremely interesting and important results have been obtained by Mr. Hamilton, the well-known superintendent for the Clyde Shipping Company, and the author is indebted to him for permission to embody these results in this paper.

(1) The first steamer fitted by Mr. Hamilton was the S.S. *Lizard*, and the arrangement adopted is shown in Fig. 2 (Plate XX.), from which it will be seen that the shaft, which was iron, was run in a white metal bush, and that a long stuffing box provided at the after end was filled with white metal packing, with a turn or two of soft packing. The gland was screwed hard up solid on to the end of the bush. The space between the tube and the shaft was filled with tallow, and a pump was provided in the tunnel for forcing oil into the tube as required. A peculiarity about this arrangement is the large overhang of the propeller, which was subsequently recognised as a bad feature, and reduced.

The following tabular statement shows the results obtained from this arrangement :—

Built and fitted ...	September, 1895.
Dry-docked	January, 1897. Shaft down $\frac{1}{8}$ in. full.
"	January, 1898. Shaft down $\frac{3}{16}$ in. full.
"	January, 1899. Shaft drawn, white metal bush renewed, overhang of propeller reduced. The shaft showed up rather rough and reedy, owing to the action of the salt water which had got in. It was also badly wasted next the propeller.
"	February, 1900. Shaft down $\frac{3}{16}$ in. full.
"	January, 1901. New propeller shaft fitted, having brass after liner 6 ft. long, lower half of bearing white metal, top half wood ; also after stuffing box reduced by about 6 in.

At every docking, melted tallow was run into the tube, the amount required varying from a few pounds, when the shaft was seen in place, up to about 120 lbs., when drawn in for examination.

(2) S.S. *Garmoyle.*—The arrangement in this case is substantially the same as that shown in Fig. 2 for the *Lizard.* There is a slight alteration in the white metal bush, but the arrangement of the after stuffing box, overhang of propeller, &c., are the same. A continuous supply of oil was pumped into the tube by a force pump worked with a ratchet off the shaft in the tunnel.

Built and fitted ...	January, 1896.
Dry-docked	May, 1897. Shaft down $\frac{1}{8}$ in. full.
"	December, 1897. Shaft down $\frac{3}{16}$ in. bare.
"	January, 1899. Shaft down $\frac{3}{16}$ in.
"	November, 1899. Shaft drawn, white metal bush renewed, overhang of propeller reduced. The shaft was badly corroded at the after end next the propeller, but otherwise in good order.
"	November, 1900. Shaft down $\frac{3}{16}$ in. bare.

(3) S.S. *Skerryvore* :—

Built and fitted ...	March, 1898.
Dry-docked	December, 1898. Shaft down $\frac{3}{16}$ in. bare.
"	March, 1900. Shaft down $\frac{3}{16}$ in.
"	February, 1901. Shaft drawn, bush refilled with white metal, overhang of propeller reduced, depth of stuffing box reduced. Shaft in good order in tube, but slightly corroded next propeller.

Other steamers fitted in this way are the *Saltees* and the *Toward*, both newer than those already referred to. These boats are trading to Cork, Waterford, Southampton, &c., where they frequently have to drag through sand and mud, so that the results obtained are perhaps less satisfactory than would be the case if their trade were oversea instead of coasting.

The S.S. *Lizard* has, as already stated, been fitted with an after brass liner, and Mr. Hamilton does not intend to continue having shafts without liners, so that in due course all the others will come to be so fitted.

On analysing these results, it will be observed that:—

(1) Every shaft was corroded between the propeller boss and the after gland, as in the case of the S.S. *Carlisle* already referred to.

(2) The examination in dry dock showed the ill effects of overhang of propeller, so that the less this can be made, the better.

(3) In every case water got into the tube, showing that, in order to make this arrangement successful, the after stuffing box must be designed in such a way that the gland is either automatically adjustable or under the control of the engineers in the ship.

(4) All these shafts were made of iron, and with the action of the salt water, the reedy nature of the material showed up, and made the wearing surface somewhat rough.

The next cases are those of several steamers belonging to the Stag Line, Ltd., of North Shields, and the author is indebted to Mr. Joseph Robinson, the managing owner, for his courtesy in giving permission for these results being brought before this Institution.

The first vessel treated in this way was the S.S. *Clematis* (345 ft. × 46 ft. × 19 ft. 3 in.), built on the Tyne in July, 1898, and fitted with a linerless propeller shaft 13 in. diameter, the size required by Lloyd's rules (1901-2) being $11\frac{3}{4}$ in. The stern tube was fitted with white metal bearing strips and the annular space round the shaft filled with oil. The arrangement was somewhat similar to the S.S. *Carlisle* (Fig. 1), with the addition of Hunter & Milne's patent packing, which was fitted to exclude water from the tube. A sketch of this design is given in Figs. 3, 4, and 5 (Plate XX.). The results obtained from this shaft have been most encouraging. It was examined in dry dock about every six months for the next three years, and at the end of that time the white metal was worn down only $\frac{1}{32}$ in. The shaft was then drawn in for examination, and found to be in a most satisfactory condition. Up to December, 1901, the vessel had steamed over 135,000 miles, many of the runs being made light; also in the ordinary course of her trading, she was navigating muddy and sandy rivers. The owners looked upon the result of this experiment as encouraging enough to warrant them in giving further trials, and accordingly two other steamers (built in December, 1899, and February, 1900) were fitted in the same way, and have, up to the present time, given entire satisfaction.

The S.S. *Gloxinia* (313 ft. × 45 ft. × 23 ft.), belonging to the same owners, was fitted with the usual shaft, having two separate liners, in December, 1896. At the end of four years (during which time one shaft broke at the after end of the after liner, and another was condemned) the vessel was fitted with a parallel shaft running on a white metal bush, and fitted with Hunter & Milne's patent arrangement already referred to. The original shaft was $11\frac{1}{2}$ in. in diameter, and the new one $12\frac{7}{8}$ in., being the diameter over the liners of the old shaft. This steamer has since then given every satisfaction, and no further trouble is anticipated with the propeller shaft. This arrangement is now being supplied to other steamers, and can be fitted to any existing stern tube, with only some trifling alterations.

The attention of the meeting is now directed to a modified form of stern tube, having an improved bearing surface for, and provided with efficient means of lubricating the propeller shaft. (See Figs. 8 to 13, Plate XX.)

The tube itself is made of cast steel, and bored out at each end to receive brass bushes filled with either white metal or lignum vitæ staves. The bush at the forward end forms a shallow stuffing box for holding oil in the tube. Along the body of the tube are cast four pockets, extending from the after peak bulkhead to the eye of the stern post, in which are fitted four long bolts. The after ends of these bolts are screwed, and work in brass nuts let into recesses cast in the end of the stern tube, as shown in Fig. 13. The forward ends extend through suitable oil-tight glands in the flange of the tube into the tunnel recess, where they are under the control of the engineers, who can screw them up as may be necessary. An outside stuffing box is fastened to the nut of the stern tube, and a solid gun-metal flanged ring is secured water-tight to the propeller, concentric with the shaft. These together form a recess for two turns of special packing compressed by a gland which can be advanced by screwing up the long bolts already referred to.

It is proposed to make this packing of hollow square rubber, coated with specially prepared canvas. The hollow core would have sealed ends, so that when fitted in place, the imprisoned air would form a spring always tending to keep the surface of the packing pressed against the revolving ring on the propeller. In this way, it is thought, the gland would keep practically tight even with the small amount of slackness in the bearing necessary for good working. The arrangement as proposed permits of the gland being readily packed by simply removing the stuffing box, which is made in halves for the purpose.

The annular space in the tube round the shaft is to be filled with oil drawn from a tank secured in some convenient position, either in the tunnel recess or on the after peak bulkhead.

It is thought that with the arrangement as given, this tank might be fitted with a gauge glass for reading the level, and secured in the tunnel recess, as high above the stern tube as practicable. The supply of oil could then be regulated by the engineer, who could draw off a quantity from the tube by the drain cock and return it to the tank. When drawing off the oil, if water was found to be present in any quantity, the outer gland would be screwed up until the flow of water was stopped. Any water then remaining would be drained off, and its place taken by a further supply of oil.

It is not anticipated that the oil consumption would be a serious item, as the amount of leakage should be small, and any leakage at all could at once be ascertained by examining the level in the oil tank, or drawing off a quantity from the tube.

If preferred, an oil pump could be fitted in conjunction with the filling-pipe, so that by forcing in oil every part of the bearing is bound to be supplied. This, however, would hardly be necessary with the arrangement as given.

The several sketches (Figs. 8 to 13, Plate XX.) make the arrangement clear, though necessarily some small details have been omitted. The design, it will be noticed, is for one of the short stern tubes which are frequently fitted in steamers with pretty full lines, but it is, of course, equally well adapted for the finer boats with long tubes.

A variation in this design is one in which the boss of the propeller is cored out to receive the end of the bearing, and in this way a further reduction in overhang of the propeller is obtained. Rough sketches showing this arrangement are given in Figs. 14 to 16 (Plate XXI.), though the details have not been fully worked out.

The advantages of this design over existing practice are :—

(1) Reduced stress in the propeller shaft by the extension of the bearing closer up to the propeller.

(2) Prevention of local action by abolishing the brass liners, and thus preserving continuity in the strength of the shaft.

(3) More efficient lubrication of the propeller shaft, which revolves in a bath of oil on a metal bearing, and is thus free from excessive slackness, or wear down out of line.

(4) The after gland is adjustable from within the vessel, so that any leakage of oil can be detected and remedied. In many cases also, this gland could be re-packed without dry-docking, *i.e.*, by simply tipping the steamer.

The experience of the last few years seems to show that the use of separate liners on propeller shafts is a bad practice. The reasons why this is so have been already referred to, and were given at length in the author's paper of 1900.

The practice of fitting a continuous liner is now becoming more common, and is certainly an improvement, though there are several objections to it, which should not be overlooked.

These objections are :—

(1) Continuity in the strength of the shaft is not maintained. The weakest section is just forward of the propeller boss, at a part ill fitted to sustain a sudden shock such as would be occasioned by the propeller striking any floating object, or even by the violent racing of the engines. It is believed that many of the cases in which propellers are lost by fracture at this point are to be explained in this way.

(2) There is difficulty in making a thoroughly watertight joint between the liner and the propeller, even when a rubber ring is fitted, as is usually done. The writer has examined and condemned many shafts on account of defects at this part occasioned by water getting in and causing the grooving action usually found at any plane of weakness.

(3) The body of the shaft cannot be examined after the liner has once been fitted, so that any latent defect in the material would not be discovered in time to prevent a breakdown.

(4) There is also the greatly increased first cost, though this would not be considered excessive, provided the cure were effective.

The adoption of linerless shafts gets rid of all these objections at once, though it introduces other difficulties which were referred to in the early part of this paper. It was shown that these difficulties had been successfully overcome by the use of Hunter & Milne's packing gland at the after end, and there seems no reason why this arrangement should not work well in many cases.

Another device for answering the same purpose is Cedervale's protective lubricating box, which is fitted in a similar manner to the one just referred to, and is illustrated in Figs. 6 and 7 (Plate XX.). It is well known that the result of this has been in some cases very satisfactory, though it must be admitted that the reports are somewhat conflicting ; and in several instances, notably those of the larger steamers, much trouble and expense have been caused to owners by the failure of the arrangement.

It will be seen from the illustrations that the design of Hunter & Milne is to a certain extent similar to that of Cedervale, and is, in the writer's opinion, open to the same objections.

The defects common to both appear to be:—

(1) There are a considerable number of small working parts which require to be carefully fitted together, and which are liable to give trouble in coupling up the propeller, as this work is frequently done in a great hurry.

(2) The springs which supply the pressure of the rubbing surface are not easily adjusted without docking the steamer, and the packing rings cannot be removed until either the propeller is taken away, or the shaft is disconnected in the tunnel and drawn aft several inches.

(3) The varying length of the shafting due to changes of temperature must cause considerable variation in the pressure exerted by the springs, and be likely to cause leakage. The writer has recently had some observations taken with a view to estimating this amount. Instructions were given to the engineers of several steamers to keep a log recording the temperatures of the sea, of the air in tunnel, and of the oil in the caps of the tunnel bearings. The air in the tunnel was measured at each end and in the middle, and the mean figure taken. It seems reasonable to assume that the shell of the ship under water would take the temperature of the sea; while the temperature of the shaft would not be less than that of the air in the tunnel, and in some cases would considerably exceed it, owing to the bearings running with a working heat. With these assumptions, it would follow that shafting 150 ft. from the low-pressure engine to the propeller shaft coupling would alter in length from $\frac{1}{4}$ in. to $\frac{1}{2}$ in., due to an increase of temperature of the shafting over the ship of from 20° to 40°.

The result of the observations confirms this reasoning. It was found that the temperature of the air in the tunnel was 13° to 25° in excess of the sea, and the tunnel bearings were hotter than the air by from 12° to 75°. In this last case, the elongation was measured by a gauge (Fig. 17, Plate XXI.) made in port with everything cool, and on trying it later with the bearings working warm, it showed that the shaft had lengthened $\frac{3}{8}$ in. The length of the shaft from the low-pressure engine to the propeller shaft coupling was 105 ft., so that, in a large ship, the elongation might reach $\frac{1}{2}$ in. This is not a large amount, but it would materially affect the satisfactory working of springs 2 in. to 3 in. long. It has been suggested that some instances of the failure of Cedervale's arrangement were due to this cause.

In the design shown in Figs. 18 to 20 (Plate XXI.), these objections have been overcome. The number of working parts has been considerably reduced, the packing can be examined or renewed without disturbing the propeller or shaft coupling, and the fore-and-aft movement of the shaft cannot affect its efficiency.

It will be seen that a brass collar ring is threaded on the shaft and bolted watertight to the propeller, forming a smooth working surface for the packing, and protecting this part of the shaft from the action of the water. The packing box or casing is formed of two rings, as shown, the after one of which is made in halves for convenience of removal. This casing encloses a rubber tube, which presses the packing against the collar, thus forming a watertight joint. The packing consists of two gun-metal rings, made in halves, and arranged so as to break joint.

This design is the result of many experiments with different tubes and forms of packing ring made during the last year. A sketch of the experimental apparatus is given in Fig. 21 (Plate XXI.), which sufficiently explains itself. The shaft was 10 in. in diameter, and revolved at about 60 revolutions per minute. It will be observed that the rubber tube was fitted with an air valve and pressure gauge, which permitted the pressure to be controlled and varied at will. A pressure of 10 lbs. was found to be sufficient to keep the gland absolutely tight.

The simplest form of this packing is a thick soft rubber tube with sealed ends (Figs. 18 to 20, Plate XXI.), having just sufficient compression to keep the rings close against the collar. In most cases, this would be quite sufficient for all practical purposes. In other cases, where it might be advisable to retain the means of inflating the tube, the air valve would, of course, be fitted in some place accessible to the engineer, who could vary the pressure as found necessary.

This design can readily be fitted to any existing steamer, and would be specially useful in the case of twin screw, or turbine steamers, with outboard shafting supported on "A" brackets. In turbine steamers, where the shafts revolve at a high rate of speed, some such arrangement would allow the bearings to be properly lubricated, and thus reduce tear and wear.

As the result of his experience with propeller shafts, the author would suggest the following improvements in design as offering in the meantime, perhaps, the best available solution of this problem :—

Material.—Ingot steel hydraulically forged, and without liners.

Diameter.—To be equal to an ordinary shaft over the liners, *i.e.*, about 1½ in. over rule size.

Bearing.—To be white metal and extended as close as possible to the propeller, so that the overhang may be a minimum.

Lubrication.—The shaft to be fitted with some form of after gland, so that it may run in a bath of oil and thus prevent wear down.

If these suggestions were generally adopted, it is anticipated that the life of propeller shafts would be greatly extended and the present lamentable list of shaft failures considerably curtailed.

In addition to the names already mentioned in the paper, the author has to express his acknowledgments to Messrs. John McNeil & Co., of Glasgow, who kindly carried out all the experimental work referred to.

DISCUSSION.

Mr. H. M. Rounthwaite (Member): Mr. Chairman and Gentlemen, it is difficult to over-estimate the importance of this question at which Mr. Younger keeps pegging away. I am very glad to see that he does so. It is a matter of the very greatest importance to all connected with steamships. There are, however, one or two points I have noted and which I should like to discuss. The first is that, to my knowledge, naked propeller shafts—that is, shafts without sleeves of any description—have been in use for at least thirty years. Steamers have been running out of the Humber to the ports of North Europe with linerless shafts and white metal bearings for fully that length of time. The fact that they have continued so to run, shows that the results have been fairly satisfactory. But it is important to note that the white metal used is not what might be called a copper white metal. It is not a Babbett metal, that is, a metal containing eight or nine per cent. of copper and possibly the rest tin, and no zinc, so that the copper is left practically free to attack the shaft; it is more what might be called a Fenton's metal. It is a zinc white metal containing only about one per cent. of copper and something like eighty per cent. of zinc. I may incidentally mention that white metals with a lead basis have also been found objectionable, because they corrode the shafts. There are some well-known white metals on the market that are composed principally of lead, I believe. The results obtained from the *Clematis*, and given by Mr. Younger, are remarkably good, and would seem difficult to beat; but there are objections, as the author has pointed out, to the form of apparatus used. As regards the stern tubes shown in Figs. 8 to 13 (Plate XX.), I do not like the rubber: "rubber packing covered with canvas" is the description in the paper. Of course, if oil can be entirely excluded, it will do very well, but I doubt whether oil can be excluded, and, of course, when oil and rubber come together, the oil very soon destroys the rubber. I have some doubt, too, about the stuffing boxes, and the method of tightening up. In order that the stuffing box may be in halves as he sketches it, and removable, it would have to be attached by means of screws instead of by square-necked studs, and there is always risk in using screws. In fact, it requires actual experience to convince one of what will come adrift about a propeller; everything comes adrift; nothing seems to stand the vibration. At the middle of page 162, Mr. Younger incidentally speaks of the short stern tubes of full vessels. I think that, probably, some of the mischief which has been occurring with propeller shafts is due to these very short stern tubes. One-eighth inch wear down at the outer end in an 8 ft. stern tube would, of course, give double the inclination of shaft, and double the stress, that it would give in a 16 ft. stern tube; and I do not know that our registration societies have taken that into account. I speak of the distance between the after bush of the stern tube and the after tunnel bearing. I take it that, in very full ships, this distance is very short and that the shaft is not so well guided as it would be with a longer tube. There is no difficulty in making the tube longer even in a very full ship. I see no reason why there should not be a partial after collision bulkhead placed in the tunnel, why the after part of the tunnel (say eight or ten feet) should not form part of the after peak space. In the design shown in Figs. 18 to 20 (Plate XXI.), the last one that the author described, there is again the objectionable indiarubber, which I am afraid would give trouble. I have taken the liberty of sketching out a suggestion here (Fig. 1), which, I think, would be, in some respects, simple, and would get over some of the difficulties I have referred to. I am proposing to have at A a ring with half lapped joints fitted with tongue pieces, and held in contact with the brass sleeve B by means of a spiral spring (9 or 10 in. long) fitted into a groove on the outer circumference with a piece of $\frac{5}{16}$ in. flexible wire rope carried round to complete the circuit. It is not necessary to make the spring the full length required to pass round the whole circumference.

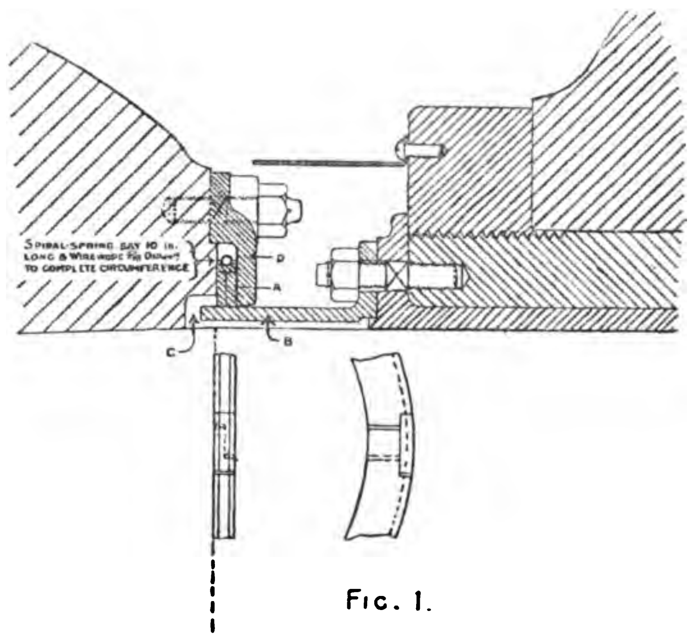

FIG. 1.

I think, in some respects, this would be simpler. The space C would be full of oil, and as the supply tank would be above the water line, the oil would tend to press the ring against the outer cover D. The oil would also pass up into the spring space and prevent corrosion of the spring. In the case of a twin-screw ship, instead of the propeller boss here shown, there would simply be a cast-iron or cast steel disc, keyed on to the shaft, immediately aft of the stern tube, and another similar one at the forward end of the bearing in the after bracket or spectacle frame. There is just one other point which concerns both this design and that shown in Mr. Younger's paper. It would not be possible to keep the oil out of the propeller boss, as it would pass in down the feather-way. I think it would be a good thing to revive an old practice, and to fill the propeller boss solid with tallow. The ring which I have shown of brass or gun-metal may be of vulcanite, as this material has been found to work very well with gun-metal. The lignum-vitæ bearing has been a good servant for many years, but I think it will have to go before we are much older.

Mr. S. W. BARNABY (Member of Council): We are very grateful to Mr. Scott Younger for the two papers he has given us on this subject. Many of the members must remember his last paper, where he showed that many of these accidents to shaf s were accounted for probably by the propellers being run for long voyages partly out of water, and the suggestion which he makes to-night is, I think, a much more practical one than that which he brought before the meeting last time. I was told by a foreign engineer only two or three days ago that this practice of running the shaft in a tube filled with oil was having extremely good results abroad, and he was surprised that it was not more followed in this country than it is. As to the author's proposal to extend the after bearing into the propeller boss, I think that looks like a step in the right direction.

The CHAIRMAN (Mr. A. F. Yarrow, Vice-President): I should like to ask Mr. Younger one or two questions. He says that more frequent inspection of shafts would be a very good thing, and that registration societies are recognising the necessity of it. I should like to know if inspection always indicates whether a shaft is fatigued or not. Of course, I can quite understand that if there is a flaw in the shaft, or if the shaft begins to show any signs of fracture, inspection would be a very good thing, but I should also imagine that the shaft might get fatigued, and that inspection would not show it. Shafts wear down at the outer bearing, and I imagine that one cause of shafts breaking is due to the torsional strain on the shaft, combined with its not being in line when it is worn down. Now assume, for example, that a shaft wears down, we will say, at the stern, $\frac{1}{4}$ in. It must be admitted that anything of the kind is very severe on the shaft. Would it not be possible to start the shaft $\frac{1}{8}$ in. up out of line and let it wear down $\frac{1}{4}$ in., and then it would only be $\frac{1}{8}$ in. out of line and it would not be strained so much. It would be very easy to do that, because we could bore out the stern tube and put the bush in and let the bush be bored out a little eccentric, and pushed into its place. On some locomotives, a frequent practice is to bore the crank pin, and put a bolt through the hole. The object is that if the shaft breaks at the crank pin, the bolt holds it together for a certain time, and so avoids a serious accident. In propeller shafts, it is a very common plan to bore a big hole through them, perhaps nearly half the diameter of the shaft. We sometimes hear of screw shafts breaking, of propellers getting adrift, and fouling the rudder and carrying it away. Now might it not be possible to make use of that hole through the shaft by putting a smaller shaft through it? That inner shaft would not get fatigued when the other one does, on account of its smaller diameter, and it might, in case of a fracture of the outer shaft, possibly hold on and avoid a more serious accident. It would, at any rate, keep the parts together for the time being. The hole is generally there in all large steamers, and, if it is a feasible scheme, it is a pity to throw away the opportunity of turning the hole to useful account. I should like to know from the author whether he considers that to be practicable, and whether it would offer any advantages?

Mr. MACFARLANE GRAY (Member of Council): I am sorry Mr. Manuel, of the P. and O. Line, is not here. Speaking to me on this subject a few days ago, he said that they had not had a shaft failure in all their large fleet since 1881. Perhaps someone present can tell us how that excellent result has been obtained?

Mr. EDWARDS (Member): I have, in some cases, had the shafts lined up as Mr. Yarrow suggested above the true line, so that when they had worn slightly down, they then became true. Then, when we measure the wear, we know that the real amount the shaft is out of truth is less than its appearance conveys.

Mr. A. SCOTT YOUNGER, B.Sc. (Member): At this period of the evening, Gentlemen, I feel I shall best consult your wishes by making my reply as brief as possible. I am much indebted to Mr. Rounthwaite for the remarks he made. Of course, the practice of running shafts without liners is, as he stated, a very old one, although it is really only recently that we have begun to accumulate data on the subject. Mr. Rounthwaite seems to be rather afraid of the action of the oil on the rubber packing. There is no doubt that if oil does come in contact with the rubber, it is liable to set up a somewhat destructive action. Exactly what that action is, I do not think we have enough information to tell. I am now making some experiments with a view to getting data on the matter, but they are at present very incomplete, or I would be glad to give the results. So far as they have gone, it has been shown that different oils have widely different results in their action on rubber. Paraffin was found to be absolutely destructive. Rubber, when immersed in paraffin, swelled up in the course of a few days to twice its size, and ultimately appeared almost to dissolve. The heavier oils, such as are used for lubrication, had not the same destructive effect. Any effect they have appears to be in the direction of swelling the rubber, so that, in the designs shown here, the action of the oil would tend, if anything, to make the joint tighter. In the event of the ship being docked and the tube being found damaged in any way by the action of the oil, it would be a very simple matter to fit a spare tube in its place. The cost of these tubes is very small; they are much cheaper than many of the packings that are at present in use, and one or two spare ones would usually be carried. Mr. Barnaby, in his remarks, reported good results from oil lubrication. I know several steamers that have given very satisfactory results. Some, however, which were fitted with Cedervale's patent have only done well for a time. Others, again, have given extremely unsatisfactory results, and have led their owners into enormous expense. What the reason was, I have tried to indicate in my paper. It would appear to be due to the elongation of the shaft gradually reducing the compression of the springs, and allowing the water to leak in and the oil to leak out of the after gland. Mr. Yarrow asked about more frequent inspection, and inquired whether it was always possible to infer from the examination whether a shaft was strained or not. That is an extremely difficult question to answer with any satisfaction. There is the case of the two steamers H and I referred to in the Table. These boats were sister ships, built about the same time, and their shafts were surveyed after being the same length of time at work. One shaft was passed, but broke before the next survey came round; the other appeared to be in a rather unsatisfactory condition, and was condemned. From the fact that one of these shafts broke, I think it is likely that there may have been some defect that could not be seen. That rather bears out the Chairman's suggestion that there are defects existing in a shaft which it is not always possible to detect on inspection. I believe, however, that a more frequent examination would certainly tend to reduce the number of casualties, because the action which ultimately leads to the fracture of the shaft begins almost exactly at the end of the liners, and is usually distinct enough to enable the surveyor to insert the point of his knife; it is almost like a fine saw-cut running right round the shaft. If the shaft is on the small side, this action begins soon after it has been put into regular work, and it extends very quickly. Mr. Edwards answered the Chairman's other point with regard to the advisability of starting operations with the after bearing high. The boat referred to in the paper, the s.s. *Carlisle*, was fitted with a shaft $\frac{1}{4}$ in. high, so that by the time it had worn down $\frac{1}{4}$ in. it was exactly in line. One objection to running a shaft which is down at all, is the fact that when the engine is racing, there is a certain amount of whipping action in the shaft. This is bound to jar on the bearings and set up stresses in the material of the shaft, the effect of which is certain to be serious. The Chairman's other suggestion about having a hollow tail end shaft and a long bolt inserted in it is not altogether new. I think Lloyd's chief surveyor in Glasgow, Mr. Mollison, has worked out that idea. I do not know whether he has patented it, but I have seen a design of his embodying the idea. Mr. Macfarlane Gray's reference to Mr. Manuel's experience hardly applies, I think, to the type of boat which is dealt with in this paper. The steamers which Mr. Manuel has under his charge are of such a kind that their shafts are not likely to be subjected to the ill-treatment which the ordinary cargo boat meets with daily. I fancy that their immunity from breakdowns can be explained partly in that way and partly by the fact that most of his steamers have large engines, with shafts heavy even in proportion to the engine.

The CHAIRMAN (Mr. A. F. Yarrow, Vice-President): Gentlemen, I am sure you will wish to return your best thanks to the author for this very practical and interesting paper.

RECENT SCIENTIFIC DEVELOPMENTS AND THE FUTURE OF NAVAL WARFARE.

By WILLIAM LAIRD CLOWES, Esq., Associate.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 21, 1902; the Right Hon. the Earl of GLASGOW, G.C.M.G., LL.D., President, in the Chair.]

AT a time like the present, which is one of extremely rapid scientific progress, it is especially incumbent upon us not to neglect, even for a single unnecessary day, any device which may possibly enable us, either in peace or in war, to defeat our rivals by honourable means. Scientific discovery tends ever more and more to obliterate the significance of those physical and moral differences which anciently rendered one race superior to another; brain and thought are already more potent factors in the world than mere muscles and animal courage. Moreover, we know from experience that to-day or to-morrow may witness a complete revolution of method in almost any of our processes. We ought, therefore, never to sleep, save, as it were, with our ears and eyes open.

Yet, strange to say, the amount of practical attention which we give to new machinery and appliances is often, I am afraid, inversely proportionate to the novelty and ingenuity of the device in question.

We are too prone, when examining new inventions, to admire the cleverness displayed in them, and then to reject them, wholly and finally, for one of two reasons, both of which are, in reality, quite inadequate.

One reason frequently alleged is that the invention has yet to be brought to absolute perfection, and that, pending its complete evolution, we may safely neglect it. The other reason is that the machinery or apparatus is too delicate or complicated for use by the class of workmen who are accustomed to handle the appliances which the new apparatus would supersede.

The evils resulting from such an attitude, which is characteristically a national one, are twofold. On the one hand we snub and starve the inventor, and possibly drive him elsewhere in disgust; on the other, we make the far more dangerous mistake of assuming that the tide of technical education will not rise elsewhere so long as we choose to batten it down in our own little hold.

What we ought to do is, surely, in the one case to take up promising inventions, and, turning them over for development to the brightest intellects at our command, to enjoy the exclusive profits of them as soon as they are practically perfect; and, in the other, to educate our men up to the point of being able to use delicate and complicated appliances, instead of rejecting the appliances because our existing men are incapable of handling them.

It is absurd for us to say, as in practice we do: "Don't offer us any unfamiliar novelty that isn't approximately perfected; and, above all, don't offer us any perfected novelty that isn't approximately familiar." While we continue to follow that policy, we run risk of falling out of station with the rest of the world, and of not discovering our error of position until too late.

It is mainly with the object of appealing for readier official interest in certain recent inventions, and for greater official anxiety to educate the workman up to the level of the tools already available, or about to become available, to his hand, that I venture to address you on "Some Recent Scientific Developments and the Future of Naval Warfare."

These developments must inevitably influence naval warfare strategically as well as tactically, and it is hard to say in which direction the effects are likely to be the more important.

Within the compass of little more than a lifetime, the practice of naval strategy has already been revolutionised by the introduction of steam and the electric telegraph. The principle of strategy, however, has suffered no change. It is very simple, and it may still, as in the days of Nelson, be thus formulated:—To have at the right spot, and at the right moment, a fighting force superior in *personnel*, as well as in *matériel*, to the force of the enemy at the same time and place. But the practice is still changing rapidly, and, under the influence of recent invention, it must change still more.

We have not yet realised to the full the strategical value of speed* as a factor in the successful carrying out of the fundamental object of strategy. Speed, in the present, is all that, and more than, the weather gage was in the past; and, if we neglect it, we shall cripple the hands of our admirals, no matter how many ships and men we may place at their disposal. It is the soul of all effective combination for offence; and I

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* *Engineer*, October 26, 1900; *Army and Navy Journal*, November 10, 1900; *Scientific American*, November 24, 1900; Journal of R.U.S.I., November, 1900; *Morskoi Sbornik*, May, 1900; *Mittheilung*, February, 1901; *Steamship*, March, 1901.

am not sure that it is not equally valuable as a means of defence against certain weapons which at present cannot easily be otherwise avoided, to wit, submarines. The submarine, of which more anon, is essentially a slow craft, whether she travel on the surface or below it. A large ship can have no more secure protection against the submarine than the fact that she is in very rapid motion. A submarine must come to the surface to look about her; and if her big enemy be seen to be changing position rapidly, the submarine can gather little information that is likely to be of use to her.

And here I should like to say that our own preparations for attacking submarines with spar torpedoes,* fitted to torpedo boats or destroyers, are exciting the ridicule of those foreign nations which, from experience with them, know what submarines are like. We claim that our specially rigged spar torpedo can reach a submarine at a depth of 10 ft. below the surface. Commander W. W. Kimball, U.S.N., says,† very justly: "Why a submarine should run at ten instead of thirty or forty feet does not appear; nor does it appear how the destroyer could, when the submarine showed for a few seconds, head for her, and strike her with the spar torpedo before she attained a safe depth. While the battleship, protected by the destroyer, is the proper quarry of the submarine, there seems to be no law against sinking the destroyer in passing, if her presence were inconvenient." The truth seems to be that if the submarine can be reached at all by the spar torpedo, she can, at least in the vast majority of cases, be reached much more expeditiously and certainly by means of the gun; though it may be desirable to mount guns in a special manner in order to deal with her.

The offensive usefulness of speed has, I believe, been doubled or trebled by recent improvements in wireless telegraphy.‡ It looks as if every ship, large or small, in future naval warfare might be, as it were, the mobile terminus of an unlimited number of aerial cables communicating not only with the base on shore, but also with all friendly ships within a radius of several hundreds of miles.

It is true that last year, when wireless telegraphy was employed by one side during the naval manœuvres, the system broke down, for the reason that the rival commander was able to tap the messages, and did not use wireless telegraphy himself. But the breakdown on that occasion was due entirely to the manner in which wireless telegraphy was misused. The defeated commander might either have employed a code for the

* Proceedings of U.S.N.I. (Notes), June and December, 1901; *Engineer*, May 10, 1901.

† Proceedings of U.S.N.I., December, 1901.

‡ *A. and N. Reg.*, March 17, 1900; *Elec. Rev.*, March 14, 1900; *Iron Age*, August 23, 1900; *Sc. Amer.*, September 8, 1900; *Sc. Amer.*, October 20 and November 10, 1900; *Steamship*, November, 1900; *Sc. Amer.*, March 9 and March 30, 1901; *Baltimore American*, May 8, 1901; *A. and N. J.*, May 25, 1901; *A. and N. Reg.*, April 20, May 11, and June 1, 1901; *Yacht*, July 6, 1901.

sending of his messages, or have used some variety of wireless telegraphy which was not tapable. It is merely a question, on the one hand, of a special cypher, or, on the other, of special discharge terminals, coherers, and relays.* The possible variety is infinite; and it is hardly conceivable that in war time the cypher of one side should be known to the other, or that both sides should use exactly similar instruments, similarly "attuned."

Given a good and untapable system of wireless telegraphy, utilisable over long distances at sea, naval strategy must, barring accidents, rapidly become almost an exact science. But it is desirable to have in reserve some alternative to the lofty spars which at present seem to be favoured by Mr. Marconi. They cannot be carried by any vessel in very heavy weather; and they cannot be carried by a small craft, such as a torpedo boat or a steam launch, in any weather at all. It ought to be possible to substitute for them, when necessary, kites,† such as, I believe, were successfully used by General Baden-Powell at Mafeking. These have much greater radial "command" than can be given to spars on board ship; and they can be flown by craft of all sizes. Electrical kite-flying, and wireless telegraphy by that means, should form part of the ordinary routine of every ship of war. Nor must it be forgotten that there is yet much to be learnt with regard to kites, especially kites large enough to carry with them a small motor controllable from below.

We have seen in South Africa that, assuming good intelligence to be at the disposal of a belligerent, the essence of effective practical strategy is extreme mobility —that is, extreme speed. I rejoice, therefore, though it is perhaps barely a straw to show which way the wind blows, that the Admiralty has decided to continue its experiments with turbine propellers in destroyers, and also to apply the turbine principle of propulsion to one comparatively large vessel, a third-class cruiser. I trust, too, that Whitehall has already devoted its attention to Mr. J. T. Marshall's new valve gear, for there can be no doubt that we must witness, within the next few years, an enormous increase in the speed of large fighting ships; and, then, woe betide the Power which lags behind its rivals in the matter of rapid mobility. It will see itself condemned to forego strategy altogether in its naval combinations; in other words, it will find itself confined to the local defensive.

As regards tactical factors, what I have said as to our habitual attitude to new inventions applies more perhaps to them than to strategical factors. Take, for example, the question of range-finding‡ in action at sea. I was once in one of His

* *Engineering*, May 17, 1901.

† Journal of U.S. Artill., November and December, 1900; *Sc. Amer.*, October 13, 1900.

‡ Proceedings U.S.N.I., December, 1900; *Scientific American*, November 17, 1900; Journal R.U.S.I., September, 1900; *Army and Navy Journal*, January 19, 1901; Proceedings U.S.N.I. (notes), June, 1901; *Engineer*, March 29, 1901; Proceedings U.S.N.I., September, 1901.

Majesty's ships in which a set of range-finding instruments was fitted experimentally. The appliances were rejected without undergoing a really fair trial. They were tried superficially by people who knew next to nothing about them; and they were condemned, not, so far as I could ascertain, because they were ineffective, but because they were complicated, and could not, of course, be worked easily by the untrained intelligence which was available to handle them. Surely it would have been better to adopt the invention, which had been well tried abroad, or some superior one, and to train the necessary intelligence.

In the meantime, how does our Navy hope to find the range in action? It depends mainly upon being able to note the drop of tentative shots.* Where two single ships are engaged, this method, though slow and unscientific, may possibly work; but when whole fleets are engaged, how can any human eye feel certain whence comes the particular shot the drop of which is noted. At Santiago it was found to be impossible to form any correct conclusions in that way. Now there are many excellent and almost perfect range-finders. I will mention only the Barr and Stroud, which is said to be responsible for the recent excellent shooting of the *Terrible*,† and the Zeiss stereoscopic,‡ each of which reduces the possibility of error to a minimum, though there are others almost equally good. For years we have been flirting with such inventions; for years we have refrained from taking them to our arms, because of our fear that we do not sufficiently understand them. They ought to have been long ago in all our fighting ships. By this time we should then have learnt to understand them. Other nations have adopted them. What will happen if we fall into a conflict with one of those other nations?

Naval gunnery will also be greatly improved so soon as two or three comparatively small problems which are now awaiting solution shall have been solved. Their solution ought not to be long delayed, if only the right kind of intellects can be persuaded to turn their attention to them. The great want of the day is, of course, an arrangement whereby it shall be possible to fire a projectile through moderately thick modern armour, and to burst it immediately in rear.§ Another great want is some new method of igniting smokeless powders.‖ These powders, especially when fired in relatively small guns, are, as their name implies, practically smokeless. But at present

* Proceedings U.S.N.I. (Notes), March, 1901.

† *Engineer*, January 25, 1901; *A. and N. G.*, June 29, 1901. ‡ "Notes on Nav. Prog.," July, 1900.

§ *Engineer*, June 1, June 15, and June 29, 1900, August 2, 1901; *Sc. Amer.*, July 21, 1900; *A. and N. J.*, June 23, 1900; *Mar. Eng.*, October, 1900; *Mittheil.*, March, 1901; Proceedings U.S.N.I. (notes), June, and (Alger) September, 1901; *Engineering*, August 9, 1901.

‖ *A. and N. Reg.*, October 20, 1900; *Sc. Amer.*, September 1, 1900; Proceedings U.S.N.I. (notes), March, 1901; *Engineering*, May 31 and June 7, 1901; July 5, 1901.

the most convenient method of igniting them is found to be through the medium of an ignition charge of black powder of the old smoky kind. In small weapons, the ignition charge and the quantity of smoke produced by it are insignificant; but in the case of heavy guns the ignition charge alone comprises about as much black powder as formerly would have sufficed for a couple of full charges for the old 64-pounder muzzle-loader; and, the volume of smoke thus produced being very considerable, the advantages of employing smokeless powder are to a large extent neutralised.

A very noteworthy development of recent science, and one that cannot fail greatly to influence the tactics of future naval warfare, is the modern submarine.* I am not a submarine enthusiast; but it is impossible not to recognise that the extension of the open field of naval operations from a space of only two dimensions to one of three is too significant to be lightly regarded. The best existing submarine is very slow, very blind, of limited radius of action, and very liable to accident; but it is vain utterly to deny the value—especially the moral value—of a craft which, without leaving your immediate vicinity, can move altogether out of your sphere of activity, and still, perhaps, deal you a fatal injury. It seems to me that the submarine, even if it be carried no further than at present, means the doom of the old-fashioned blockade.

But I am sure that the submarine will be carried very much further than at present, and that already we may see traced out before us the lines along which it is destined to develop.

The weakest points of the best existing submarines are: that they cannot see clearly unless they come to the surface to do so; that they cannot be sure of maintaining a given course under water, even by utilising the Obry apparatus †; and that the lives of the crews within them are exposed, especially in war time, to extreme risks.

During the past three or four years numerous ingenious inventors have turned their attention to this subject,‡ with a view to producing a vessel which shall be capable of moving at considerable speed beneath the surface of the water; which shall not need a human crew; which shall not want to see whither it is bound; which shall be controllable at every moment of its course; which shall not expose those who work it to extraordinary risks; and which shall be manageable from a distance without the

* *Steamship*, April and May, 1900; *U. S. G.*, May 19, 1900; *A. and N. Reg.* October 13, 1900; *Mar. Eng.*, October, 1900; *A and N. J.*, January 12, 1901; March 23, 1901; *A. and N. Reg.*, February 9, 1900; *Engineer*, January 18 and 25, February 1, March 1, and March 8, 1901; *A. and N. J.*, September 7, 1901.

† *Engineering*, April 12, 1901.

‡ *Sc. American*, May 12, 1900.

intervention of wires or other visible connections. It is sought, in a word, to combine the useful features of the existing submarine, of the automobile torpedo, of the electrical countermining launch, and of the Brennan torpedo; dispensing, at the same time, with material ties between the operator and the weapon, and securing a range which, though less than that of the submarine, shall be far greater than that of the countermining launch, the Whitehead, or the Brennan. Some scores of patents bearing upon these projects have been issued to Messrs. Axel Orling and James Tarbotton Armstrong,* Arthur A. Govan, Cecil Varicas†, and Bradley A. Fiske, the last-named being the well-known American naval officer who is famous in connection with more than one range-finder, and with other inventions designed to influence the future of naval warfare.

These gentlemen utilise various forms of energy in various ways; and it is impossible here to go into details of their inventions. It must suffice to say that, although no perfect form of vessel controllable by wireless currents—a form to which I have ventured to give the generic name of "Actinaut"—has yet been produced, more than enough has been accomplished to demonstrate that what it is sought to effect can and will be effected in the near future. Indeed, if it were possible to induce these rival inventors to combine and co-operate, and if it were possible to place at their disposal the knowledge and experience of half a dozen men such as Lord Kelvin, Sir Hiram Maxim, Mr. Brennan, and Mr. Marconi, I verily believe that you might have the perfected engine before you on this day next year. When that perfected engine is produced, it cannot fail to work something like a revolution in naval warfare.

All these considerations bring me back again to one of the points from which I started. Our best available tools are rapidly getting beyond the effective control of our best available men; and the real lesson of the situation undoubtedly is that, if we would properly utilise all the resources which science has placed, and will presently place at our disposal, for the prosecution of naval warfare, we must greatly improve the scientific standard of the *personnel.*

It is significant that Lord Charles Beresford, without committing himself to any expression of opinion as to the merits of certain types of water-tube boilers, has hinted his belief that many of the breakdowns of those boilers may possibly be attributable, not so much to defects inherent in the boilers, as to the incompetency of the working staff; an incompetency due to lack of training and experience, and perhaps also to short-handedness.

* *New Lib. Rev.*, June, 1901; *A. and N. J.*, November 16, 1901.
† *Sc. American*, February 16, 1901.

The present Board of Admiralty is admittedly anxious to make the Naval service all that it should be; nor does it resent friendly criticism. I would therefore ask their Lordships to reflect whether the present methods of dealing with the scientific problems and daily work of the Royal Navy can possibly produce satisfactory results.

There are two categories of scientific officers in the service—the engineers proper, and the specialist executive officers. The engineers are men with a relatively long, broad, and deep scientific and technical training. They are an expensive class, and at present, the Navy has confessedly failed to attract and retain the best examples of the class.

The specialist executive officers are, so Mr. C. M. Johnson, R.N., has irreverently said,* "men who dabble in electricity, fiddle about with files and hammers, set up amateur lathes in their cabins, and imagine that they are making engineers of themselves." I do not associate myself with this description of a class of officers who, no matter what else may be said of them, are remarkably keen, and do their work astonishingly well, so far as the conditions permit. But the conditions do not permit much. A torpedo-lieutenant generally gets about seventeen months of technical training in the course of his career. This is, naturally, not enough to make a well-equipped electrical engineer of even the most brilliant of men, still less is it enough to make of him a mechanical and hydraulic engineer as well.

Nevertheless, with a view, I suppose, to economising expenses, and to restricting the total number of commissioned officers carried, the Admiralty has entrusted a great many purely engineering duties to specialist executives, and, in addition, has turned over the entire control of the engineering department in small ships to a warrant officer—an artificer-engineer. Not only is this officer of necessity a man of limited education and experience, but also he is now a man less experienced than his fellows formerly were; for a lowering of the standard of qualification has recently been sanctioned.

In the meantime, to assist the specialist executives, a class of ratings known as electrical fitters has been lately called into being. This is composed of men who have very little electrical knowledge at all.

And so we see that, whereas at one time all the engineering business of the ship was in the hands of properly qualified engineer officers, much of it is now entrusted to specialist executives, much to warrant officers, and some to people admittedly possessed of hardly any scientific training at all.

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* Discussion on Mr. D. B. Morison's paper before N.E. Coast Inst. of Engineers and Shipbuilders, February, 1902.

While, in short, the *matériel* has been improving yearly, the *personnel* has been assuming more and more the character of a penny-wise-pound-foolish makeshift.

This state of affairs must, I think, be remedied if we would profit fully by recent scientific developments. The brightest scientific intelligences ought to be attracted to the Navy, and to be retained there when once they have been engaged; and I see no reason why they should not be. We are now paying five shillings a day in South Africa to soldiers—men whose necessary qualifications are little higher than those of unskilled labourers. It can hardly be doubted that a first-rate naval engineer officer, even if you have to pay him a thousand a year, is a much cheaper article than an Imperial Yeoman at £91 5s. Moreover, while you can pick up the latter at any time, you can secure the former only if you engage and train him at a time when, as at present, you are sorely tempted to do without him, and to entrust his work to an amateur.

DISCUSSION.

Vice-Admiral C. C. P. FitzGerald (Associate): My Lord and Gentlemen, I have listened to this paper with a great deal of interest, as I am sure you all have, but, I must say, with no small amount of astonishment, because I venture to think it is rather out of date. That may sound somewhat strange as regards the nature of the paper, but it applies more particularly to the charge that we are not ready to take up inventions that are put before us. I think it is out of date in that respect. A quarter or perhaps half a century ago, when you and I, my Lord, had the honour of joining His Majesty's Navy, there was, no doubt, a very strong feeling against all inventions. Old sailors thought that they could work their ships with sails, and they did not want to be bothered with smoke-jacks, or modern innovations of any kind. I think that feeling is now dead, and that we are ready to take up everything of promise. I am answering for the Navy, because this paper is rather a charge against us personally, and I take it as rather a personal matter. I do not think you can expect us to take to our arms (as Mr. Laird Clowes says) the modern fad of every inventor that comes along with some novel idea to show us how to cat our anchors, or to lower or hoist our boats without danger. We are quite ready to look at and try anything which offers a fair prospect of success, and I do not think we can be properly accused of not doing so. We are quite ready to listen to civilians or to anybody else who gives us good advice, especially naval architects, and the progressive spirit which they have shown has done an enormous amount to help us. Now I have said that I think the subject-matter of this paper is a little out of date, and the general remarks somewhat unfair to us, but, with your permission, I should just like to mention one or two points, because it is no use merely making general statements. I would rather like to know what the author means when he uses the term "we" so very often in the second, third, fourth, and fifth paragraphs?

Mr. Laird Clowes: The country as a whole.

Admiral FitzGerald: The country generally, or the Navy, or the naval architects?

Mr. Laird Clowes: The country as a whole.

Admiral FitzGerald: Mr. Laird Clowes says "we snub and starve the inventor." All I can say is that we do not do so in the Navy, and I do not think naval architects do so. Then he tells us, a little further on, that "we have not yet realised to the full the strategical value of speed." That is news to me. Certainly, if anybody has, we have. Naval officers have always been worrying for speed at all hazards, and we certainly do realise its value to the full. This accusation does not apply to us, whatever it may do to the general public of England, though I do not think it matters much what they think of it. Again, he says that "speed is the soul of all effective combinations for offence." That, to me, sounds like a truism. Then he comes to the question of submarine boats. I do not know that I am prepared to believe in the spar torpedo, but we can let it off about ten feet under water, and it might give a submarine a nasty shock if one was anywhere near; but if Mr. Clowes can show us a new way of mounting a gun which can explode a good sized charge of gun-cotton more than ten or fifteen feet under water somewhere near the submarine, we shall be very much obliged to him. Next, he tells us of a case where he was on board one of His Majesty's ships, and they gave the officers a range-finding apparatus which was fitted experimentally, and he says it was rejected without having a fair trial. That is his own personal experience. I do not, of course, deny that he saw it, but, at all events, it is not my experience. These range-finders are things I feel most strongly about, and I have always been trying to get something to find the range with. I know it is a matter of the highest importance. By a strange coincidence, I only came back from Glasgow on Monday, and I was talking to Professor Barr about this very thing, and I pointed out to him some of the reasons why the use of his range-finder was so very limited as not to be applicable in action. By another strange coincidence, I got a letter from my friend Captain Colville, who was making experiments (this is more or less confidential) in Bermuda with the *Scorpion*, where the *Crescent* steamed up at fifteen knots with everything prepared for war, and he has pointed out some technical difficulties which I mean to send on to Professor Barr, with a view to his receiving information from a man who has been lately using the range-finder under war conditions, in order that he may perfect it and make it, if possible, more fit for use in our Navy. Now I will go to another paragraph. Mr. Clowes says, "For several years we have been flirting with such inventions; for years we have refrained from taking them to our arms, because of our fear that we do not sufficiently understand them." He says it would be all right if one could find the right kind of intellect to deal with them. I hope he will. I do not know where you are going to hunt for them if you do not find them amongst the naval architects! If we have the right kind of intellects, of course they will do it. Then we are informed that the submarine boat in its present development means the doom of the old-fashioned blockade. Well, really, the old-fashioned blockade was doomed quite a quarter of a century ago, when torpedo boats came out. Nobody has had the slightest notion of the old-fashioned blockade for twenty-five years, to my knowledge. Again, there is one little point here which I do not think quite fair to naval officers; in fact, it seems to me to be rather contradictory. He says, "Our best available tools are rapidly getting beyond the effective control of our best available men." That, at any rate, is a general indictment. And again: "We must greatly improve the scientific standard of our personnel." We are trying all we can, I think, to do that. Then, on page 178, the author quotes Mr. Johnson, although he says he does not endorse his opinion: "The specialist executive officers are men who dabble in electricity, fiddle about with files and hammers, set up amateur lathes in their cabins, and imagine that they are making engineers of themselves." He says that he does not endorse that. If so, why does he quote it? It seems to me that one of his statements contradicts the other. In one place he finds fault with the specialist officers because they do not attend to these scientific and technical matters, and, in another place, if these young men show a special aptitude for this work, and dabble in electricity, and fiddle about with files and hammers and set up lathes in their cabins, they are condemned as tinkers—that is, when they are doing their best to do the thing which he accuses them of not doing. I should like just to make one remark about this last paragraph, because I think it is very amusing. "We are now paying five shillings a day in South Africa to soldiers," and then, a little lower down, he says that we are paying £91 5s. a year to Imperial yeomen, and that an engineer is really worth £1,000. Really, I do not see the relevance of that, or what on earth the South African War has to do with the subject. I have come to the conclusion that this very discursive but interesting paper does not throw any new light on the subjects dealt with, nor do I believe it will increase or quicken the interest which naval architects take in naval science. It is, as I said, an amusing and interesting paper, but I thought it well that a naval officer should point out that we are not quite so guilty of neglect of modern inventions as this paper might lead you to believe.

Rear-Admiral W. H. Henderson (Associate): My Lord, I beg to acknowledge the fairness and, in many respects, the justness of the lecturer's criticisms, but he must remember that our intelligence in the service is a reflex of that of the nation; the percentage above or below the average intelligence is not less than that of any other class or profession. If our education is not what it ought to be, the Service, like the rest of us, feels the effect of our national education being behind the times. With regard to submarines, I agree that the spar torpedo is a cumbersome, crude, and old-fashioned weapon to attempt to meet them with. An illustration of this may be taken from the "punt hunts," which are a feature of the regattas round our coasts, where a six-oared galley is pitted against one man in a duck punt, who never need be caught. I do not suggest that the submarine is as mobile as the punt, but the destroyer is less so than the galley, and I do not think that it will ever, except by accident, get within striking distance. Besides, ships must have some means of defence, and I have always considered that high angle fire from a mortar or howitzer, using high explosives, and a shell fitted with under-water time fuses, offers much greater practicable results both for offence and defence. As to the submarines themselves, there is, on the other hand, the danger of our not recognising their limitations, and being carried away by enthusiasts, as we were in the days of the torpedo-boat mania. Wireless telegraphy has now introduced a new factor. It is not, perhaps, too far fetched an idea, to imagine that within a restricted area, such as the Mediterranean, the Commander-in-Chief may some day direct the operations of his squadrons from a fixed base such as Malta, or Gibraltar, leaving tactics to his specially selected tactical commanders, for in this matter differentiation is required. We are quite alive to the development of wireless telegraphy, and quite in the front rank of experiment. Range-finders are of a different order, and there is truth in the author's criticisms, but there are many difficulties. I think, however, that every large ship is now fitted with them. The only other point I wish to notice is that of the engineers. My opinion is that we must never separate officers or men from the material with which they have to fight and work. We must not only understand the manipulation of the guns, torpedoes, and whatever weapons or appliances we have to deal with, but we must learn to refit, repair, and maintain them; there is no want of intelligence if only it is trained for the purpose. We have carried specialisation in these matters far enough already, and I hold that it would be a great mistake to specialise these functions in one particular class. We must, as we progress, teach our people to become more and more mechanics: it is only what is taking place on shore; the development of the motor car, for instance, means that our grooms and our coachmen will be gradually turned into mechanics. We must expect these changes, and be prepared to meet them, and I maintain that there is no want of intelligence for the purpose within the Service, if only it is properly led, encouraged, and trained.

Mr. W. H. Whiting (Member): My Lord and Gentlemen, in spite of the criticisms of Admiral FitzGerald, it seems to me that of all the papers which will be printed in this year's Transactions, this one has the greatest and most general public interest. It is well to recollect what it is that has made this interest possible. During the last fifteen years, there has occurred something very like a revolution in the state of public knowledge concerning naval affairs, both as to principles and as to facts. Captain Mahan has given us those memorable statements of the great principles underlying and governing all naval warfare, and of their exemplification in history; and, on this side of the water, Lord Brassey and the able men who have followed him, have spared neither time nor strength in laboriously accumulating those facts which are now at the disposal of everybody who cares to read them. These labours have alone made possible the very existence of an intelligent public interest in naval affairs, and in this work Mr. Laird Clowes has played no inconspicuous part. For that, if for no other reason, his paper is entitled to our careful consideration. But it has a greater claim than that. He brings before us a question, which is one of the most interesting—because one of the most practical—which could be submitted for discussion. I say "a" question, because, although, as Admiral FitzGerald has amusingly said, the paper is rather discursive—indeed, if it has a fault, it is, not that it is uninteresting, but that it is positively seductive—yet there is one main question which underlies it all, namely, what is the attitude which everybody connected with the Navy, in however humble a capacity, ought to hold towards modern scientific development. That, I take it, is the main question which Mr. Laird Clowes raises, and by itself, my Lord, it is amply sufficient for a morning's discussion. Mr. Laird Clowes makes no secret of what are his own opinions on this matter. He says that we are displaying all our old national faults, that we are rejecting inventions, discouraging inventors, and having nothing to do with any of their projects unless they are fully worked out when they are presented to us. (In discussing this subject, I ought to make clear that I am speaking entirely for myself as an individual naval constructor.) He says, further, that, to make assurance of failure doubly certain, we neglect to train men who are capable of appreciating or of handling these appliances. He adds that if we persist in this course, we shall be left behind. These are good, wholesale charges, and he points out to us a more excellent way. He says that we ought, on the one hand, to search diligently for the most promising inventions; and, secondly, that we should devote the most brilliant intellects at the disposal of the Navy to bringing these inventions to perfection. The results should then be appropriated for our exclusive benefit. In a general way, my Lord, we all agree with that. The difference of opinion comes in when we proceed to details. In a general way, too, we may say that this course is followed already. There are many inventions which, as must be perfectly well known to most of us here, have been materially modified since their first introduction into the Service. Such, for example, are the Whitehead torpedo and the submerged torpedo discharge. And those of us who have been engaged for years in perfecting these inventions, even if we cannot take to ourselves those happy words of Mr. Laird Clowes about the most brilliant intellects, we can at least say that these inventions have been made fit for practical use, and that when these results were obtained, they were appropriated for many years to the exclusive benefit of the Service. But the author proposes to go a good deal further than that. He proposes to take what one may call immature inventions, and to proceed with their development on something like a wholesale scale. There, I think, he lays himself open to three lines of criticism. The first is that he assumes, it seems to me, that such work constitutes the most profitable occupation for the brilliant intellects at our disposal. I think it is quite open to doubt—I put it only as a suggestion—whether the most capable naval designers would not be better employed in ship design proper, and the ablest naval officers in learning how to handle, even more efficiently than at present, the appliances already at their disposal. In the second place, he touches on what is still more doubtful ground when he suggests that on those two factors—the perfection and elaboration of our mechanical appliances, and the scientific knowledge of the men who are to use them—will mainly depend the results of the next great naval war. History, my Lord, affords no adequate support for that contention. It certainly does not hold good for Trafalgar and Lissa, and I very much doubt whether Mr. Laird Clowes could argue fairly for such a contention from Santiago and the Yalu. I believe that in none of those cases could that argument be fairly urged. What could undoubtedly be said is that if, in the last two battles, the crews had been exchanged, the result might have been sensibly modified, and I do not think that mechanical appliances were the chief factor in bringing about the actual results. But, my Lord, the third and the most serious point on which I differ from the author is that he seems to think it is positively a desirable thing in itself to increase, almost indefinitely, the number and complexity of the mechanical appliances which we find in our ships to-day. There, I think, he is entering on very dangerous ground. There would be, in my judgment, speaking simply as a naval constructor, hardly less risk if we went entirely in the opposite direction, and tried to sweep out this complexity altogether. We need to remember, that, in the opinion of many men, it is beyond the power of a single officer to control effectively the enormous offensive power concentrated in one ship; and if that is true to-day, how much more will it be true if we proceed still further in the direction which Mr. Laird Clowes indicates. The result will be, not merely to overwhelm the commander, but to a very large extent to cripple the ship. The interesting discussion we had the other morning on torpedo boat destroyers is entirely relevant to this question. It was pointed out quite clearly, that you have only to improve here a little and there a little, and, before you know where you are, you wipe the type out of existence altogether. Now, what is true about destroyers is true, allowing for differences in type, for every class of His Majesty's ships. You have only to add improvements here and improvements there in sufficient number, and you destroy the practicability of the type: while gaining locally, you have taken from it some of its primary characteristics. I often wish we could have a great object lesson; that we could take one of these big ships, and that there might be seen going on board at one end of her a long procession of the elaborate appliances; of the improvements of infinite complexity; of the duplicated and reduplicated fittings; of the provisions against minor and subordinate and remote risks; of all those things which are intended to get the very last one per cent. of efficiency out of every portion of the ship and her equipment; and that there might be seen going on shore, from the other end, a procession of 6 in. guns. I believe that such an object-lesson would be more instructive than many a discussion. My Lord, I have been a silent member of this Institution for nearly twenty years, and I fear that I have committed the usual fault of a beginner, and have spoken too fully and perhaps too frankly: but we have reached, it seems to me, no ordinary point. At the close of a long and memorable period of naval construction, and at the opening of another, we stand, in a sense, at the parting of the ways. What the future of naval design will be, no man can tell. But this, at any rate, is certain, that it will be profoundly influenced by the general body of naval opinion, and that this, in its turn, will be largely built up by the writings of such men as Mr. Laird Clowes, and those who labour with him. We all have the same object at heart—the success of the Navy—but we naturally look at it from different points of view. Mr. Laird Clowes desires, as it seems to me, more and more elaboration and complexity. I plead, my Lord, to-day, for more simplicity. I do so, not merely on financial grounds—those grounds which have been laid before this Institution so often and so ably by my friend, Professor Biles—although those considerations go to the very bottom of things in this connection. I appeal to you on grounds which are common to us all, common to Mr. Laird Clowes, to naval officers, and to myself, namely, that we all desire to get the maximum power into the individual ship. For that end, it is essential that we should strive our very utmost after greater economy of idea, greater directness of aim, and greater simplicity of detail design. In that path, and that path only, I am convinced, does safety lie; and if this morning's discussion should tend, in however small a degree, to influence the great body of naval opinion in that direction, Mr. Laird Clowes will have added one more to his list of public services.

Admiral Sir JOHN HOPKINS, G.C.B. (Associate): My Lord, I should like to touch on a few points of this paper with regard to which, I think, attention has not been fully drawn. To begin with, I am of opinion that the paper is a very good one for discussion. We want papers of this kind put before us occasionally, because they make us *think*, and that is what we must be doing. I do not agree with Mr. Whiting that we are going to stop short at certain developments. There, I think, he is all adrift. Development is bound to go on, and what we see nowadays is that development is attaining such a position that without it we are nowhere. Now what is the whole history of our naval construction, to take that point first? It is *development*—development in every sense and on every point. I will give you a very simple instance of that. Apparently, Mr. Whiting does not like all these mechanical appliances. There was a certain case, of which I am cognisant, in which a naval officer came forward and said, "I do not get my 6 in. shot up to my 6 in. gun nearly as quickly as I can fire them away." He was told "You must put a pile of shot behind the gun, and then you can have them ready when you want them." "But," he said, "when my pile of shot is gone, what then?" Here, in fact, development was essential, the human element not being equal to passing up shot at the rate at which it was wanted, and this could only be done by machinery. Now see the result of development. An electric appliance was arranged by which, where one shot could be passed up by human labour, six could be passed up by this mechanical appliance. Opponents said, "This electrical arrangement will break down in war time, and then what will be your position?" My friend replied, "We should fall back on the human being; but he is only one-sixth as good as electricity, while it lasts." And that is true of all mechanical appliances. Of course, they are not so simple as the human element. I am old enough, and my noble friend, the Chairman, is old enough, to remember the time when all these old guns were worked with handspikes. No doubt it was a fine order of lever in those days; we were told that it was a lever of the first class. But how about the handspike now—where has it gone? It has gone where a good many other things which are out of date have gone, and if we do not live up to date in all our mechanical appliances, with due respect to Mr. Whiting, we shall go under as well. But that is by the way. I did not intend to begin in that way on a side issue. Mr. Laird Clowes very naturally says that we ought to take up promising inventions and develop them with the brightest intellects that we have at our command. I think we all feel that there are so many inventors knocking about, that if you took up all their "promising inventions"—from their point of view—you would have no time left for anything else. I have had a good deal to do with inventions in my time at the Admiralty, and Mr. Whiting and Mr. Swan and everybody else will tell you that if your door was open for inventors all day long, you would have nothing else to do but to develop their promising inventions. There was a curious case, when I was at the Admiralty, about the Brennan torpedo that looked promising, and was. Now what happened? It was taken up by the War Office, and, before anything was made of it, it cost the country £30,000. Then there was a committee meeting at the War Office to know if they should drop the Brennan, the secret of which, they admitted, must get out sooner or later, and drop also their £30,000. Their line of argument was: we have spent £30,000, and we may as well spend £100,000 and go on with the thing, otherwise we shall lose the money that has been spent already. Of course, there is a certain amount of sense in that. That was a case of taking up a thing before it was fully developed and worked out, and practically, because our own people had worked it out, they did not like to drop the money they had spent upon it. Mr. Laird Clowes says, "We have not yet realised to the full the strategical value of speed." He cannot lay that charge against me, because in season and out of season I have advocated speed. Nearly two years ago I read a paper at the United Service Institution on the slowness of speed in our second-class cruisers. As I have said before at this Institution, we have stood still in that direction for ten years. Sir William White's genius was handicapped by sickness or opposition, and he could not tackle it, though I give him the credit of being able to do so, but at all times, and in common with a great many naval officers, I have gone in for development of speed. A cruiser is no good without speed, and, in a battleship, with speed you command the situation. Lord Charles Beresford the other day told us that Sir Gerard Noel—he did not mention the name, but we all knew who it was—lost that battle in the Channel because he was not a strategist; but we know he lost it simply because he had inferior speed to his opponent, whom he was running after and could not catch. A man with superior speed chooses his own distance and his own formation, and can annihilate you piece-meal as you come up. That is what speed does for you. It had nothing to do, on the occasion mentioned, with tactical skill at all. I do not know about our preparation for attacking submarines. I have had nothing to do with spar torpedoes lately, but I do not think the submarine will be reached with a gun. I am with Admiral FitzGerald there, as we have never yet found a gun that carries two feet under water. Now we come to the Barr and Stroud range-finder. As you all know, we have been hammering away at that for years, and we have not got it perfect yet, by a long way. I do not suppose Mr. Stroud has got it perfect to his own satisfaction. There is an amusing story about that. There was a little midshipman out in China, and he was conducting a party of ladies round his ship. They came to the Barr and Stroud range-finder, and they said, "What is that, little officer?" "They call that the peace of God, for it passeth all understanding," he said; but we have all had a lot of it, and we can now find ranges to any extent. In fact, there is now a perfect knowledge of it. Then you come to the Barr and Stroud limitation, and to our friend Captain Percy Scott, who might have made use of it, but it was not good enough for him. Here was a man who had got gunnery to such a pitch that what he wanted was to hit the centre of the bull's-eye at 2,000 yards, yet with the Barr and Stroud range-finder you are allowed three per cent. of error, and that at 2,000 yards, if my figures are right, is sixty yards. Do you think that would do for Captain Percy Scott? Not a bit of it. He was the man who invented a thing called a dotter, by which every shot which was aimed along the barrel of the gun was dotted on a target, and he knew exactly what his people were about. Then again, he trained his people to the use of the telescope. We all know how difficult it is, with any motion on a ship at sea, to fix an object in the field of the telescope. But what Captain Scott did was to educate his men to use the telescope in the same way as you use your fowling-piece against a flying object, and in process of time, the men became so educated that there was nothing that they could not get in the field of the telescope and keep it there. That was the explanation of all his good shooting—it was the dotter; and when the telescope was fitted, it enabled him to get a bull's-eye at a 2,000 yards target, when the Barr and Stroud range-finder would be out of it. I am not saying anything against the range-finder, because we are not all Captain Scotts, and no doubt the range-finder serves a useful purpose. The Barr and Stroud range-finder is also fitted on nearly every fort in England, so that you cannot say that it is not generally used. Then again, Mr. Laird Clowes says:—"Naval gunnery will also be greatly improved so soon as two or three comparatively small problems which are now awaiting solution shall have been solved. Their solution ought not to be long delayed, if only the right kind of intellects can be persuaded to turn their attention to them. The great want of the day is, of course, an arrangement whereby it shall be possible to fire a projectile through moderately thick modern armour, and to burst it immediately in rear." Well, the author will be glad to hear that this has been done. I believe it is a fact that when the 9·2 gun was fired (it is an open secret or I should not have heard it) at the *Belleisle* the other day, the armour-piercing projectile penetrated six inches of Krupp armour (or, at any rate, hardened armour) and burst inside. On p. 175 he says:—"Another great want is some new method of igniting smokeless powders. These powders, especially when fired in relatively small guns, are, as their name implies, practically smokeless. But at present, the most convenient method of igniting them is found to be through the medium of an ignition charge of black powder of the old smoky kind. In small weapons, the ignition charge and the quantity of smoke produced by it, are insignificant; but in the case of heavy guns, the ignition charge alone comprises about as much black powder as formerly would have sufficed for a couple of full charges for the old 64-pounder muzzle-loader." I do not want to criticise this paper unduly, because it is an excellent paper, and, as I said, makes us all think, but where Mr. Laird Clowes has got that from, I do not know. The full charge of an old 64-pounder was, if my gunnery recollection is right, 10 lbs., and latterly 8 lbs. You multiply the latter by 2, and that is 16 lbs. Now, it requires only about 12 oz. to ignite the charge in a 13½ inch gun—that is, a 69-ton gun; so where the 16 lbs. come from, I don't know. Then, again, on p. 176, the author says: "It seems to me that the submarine, even if it be carried no further than at present, means the doom of the old-fashioned blockade." As Admiral FitzGerald remarked, that has been doomed years and years ago. No man in his senses would attempt to blockade a port: it is out of the question. What you have to do is to watch a port with a vessel so fast that, when a 23-knot well-protected cruiser comes out, you can quietly run away from it and tell the admiral that it has come out. But ports must be watched, and no blockades are possible. I do not think it is my business as a naval officer to say whether it is possible for rival inventors to combine and co-operate, but, gentlemen, I should rather like to see Lord Kelvin, Sir Hiram Maxim, Mr. Brennan, and Mr. Marconi agree on any one subject when they all met together! I think that is where the difficulty comes in, and that is where the difficulty with experts lies. One gets up and says this, that, and the other is quite right; the next man you come to says it is all wrong, whilst somebody else won't agree with either! Then Mr. Laird Clowes goes on:—"It is significant that Lord Charles Beresford, without committing himself," &c. There is no doubt that the breakdowns in many of those boilers were attributable to their being novelties in the hands of people who had no experience with them, but so it must be with everything. You have to feel that all your men must be trained to the use of whatever is put into their hands, whether it be a boiler, or an engine, or a bit of mechanism connected with electricity or anything else. We know, in our own case, that if you give a man a new rifle and send him down to the butts, where there is a good deal of complication in sighting and so on, he will not understand how to get the best out of the weapon till he has had a few dozen or a few hundred shots. That is true of everything, and we must not shut our eyes to the fact—I do not think the naval officer has shut his eyes to the fact—that the taking up of anything new is a matter of careful handling and suitable experience until one knows what he is playing with. The modern bluejacket, and the naval officers as well, are equal to anything you can demand of them, and you do not hear of anybody returning from one of the big ships, if he has been a good deal mixed up with naval officers there, without hearing that their intelligence is equal to anything. Another complaint in this paper is that something has been taken away from the engineer and given to the naval officer. That, I think, is a very good thing. The engineer, in his own particular line, stands alone; he is a trained mechanic; he is a skilled engine-driver; he has a thousand and one little points of education which the ordinary naval officer does not get, and which put him, in his own field, above the naval officer; but when you get a scientifically inclined naval officer to work at a thing, you will find that he is just as capable of taking up a big question as an engineer. Now I will put that in a simpler way. Look at our gun-producing factories. Take Elswick: There is a certain naval officer there who is one of the moving spirits in all inventions. Take Vickers, who have done so much latterly in producing good gunnery weapons and mountings. Who does a large portion of that? A naval officer. If you believe Mr. Laird Clowes, the engineer would do it a good deal better, but, on the other hand, we see these great firms taking the naval officer to their assistance and appreciating his talents. I have said enough, and I am afraid I must have wearied you, but I think I express the naval opinion, when I say that everybody is glad of criticism from any quarter, whether it is well founded or whether it is weak, so long as it gives us something to think of and discuss; and, in the future, if criticism enables us to stand on a sound foundation, so much the better for all of us.

Admiral the Hon. Sir Edward Fremantle, G.C.B., C.M.G. (Associate): I am afraid, my Lord Glasgow, that we shall have had rather too many naval officers, but this is essentially a naval officer's paper. It is one which appeals to the naval officer most directly, and I must say it is one in which I, personally, feel very deeply interested. Now I will begin with a little criticism, in which I must, to a great extent, agree with my friend Admiral FitzGerald. I am afraid my friend Mr. Laird Clowes, whom I used to know very well a good many years ago, is scarcely up-to-date. He has, unfortunately, not been in a position to go on board men-of-war recently, I believe, and I think you can hardly keep up your knowledge, at all events in an exact manner, by simply reading, although you read everything that is printed.. You must also converse with the men who are doing the work, and you must go on board the ships where the work is being done. The illustrations given by Sir John Hopkins as to what is being done by Captain Percy Scott are cases in point. Obviously, if you start from what has been done, or the position which was taken up, by the Admiralty, or by naval officers, fifteen years ago, and then apply it to the position which they are supposed to have taken up to-day, you are likely to make a mistake. Now, having said that, I am still very much in agreement with Mr. Laird Clowes on the subject of his paper generally. It is an extremely useful thing to make us *think*, as Sir John Hopkins said, and I quite agree with the author that we are not generally alive enough to new inventions, although certainly at the present day, and very recently, we have been far more alive to them than we used to be some years ago. The very illustration which was used by Sir John Hopkins with reference to the supply of ammunition to quick-firing guns, and its now being done by electricity, is a case in point. That is true, but we were about the last maritime nation who took to supplying the guns with ammunition by electricity, and certainly, as far as I know, we were very much behind several of the great maritime nations in that respect. We have often heard that the Admiralty are very hard-worked, and that their daily routine is really too heavy for them. That is perfectly true. We all know that the Board of Admiralty—the present Board, most decidedly—are very hard-working and able men, and they give their whole energy and their whole ability to the problems before them. The daily problems are very great and very numerous. This question of inventions is a very large one. As Sir John Hopkins said, there are a great number of inventors who all have invaluable recipes, and their belief is such, that if their invention was adopted they would do wonderful things. But, naturally, what happens is, that some defect is pointed out in it; it is shown that it is not so perfect as the inventor declares it to be, and a busy man throws it aside and says, "I will have nothing further to do with it." The right thing to do in many cases is to develop the invention, and I suggest we should have a committee on inventions. Everything is being done by committees now, and it is quite right that it should be. I am satisfied that if we had committees on various subjects, constantly sitting—subjects connected with the development of the Navy—they would be found to be extremely useful. We all know that we have an Ordnance Select Committee which we used not to have. It has now become part of our routine, part of our constitution, and I am quite sure that neither the most extreme Radical nor the most extreme Conservative would ever think of abolishing it now. Why should not we have other permanent committees, and among them a committee on inventions? I should not have spoken at all, but for this question of speed having been distinctly alluded to by Mr. Laird Clowes. I had the pleasure, some years ago, to read a paper at the United Service Institution on this very subject. It was called "Speed as a Factor in Naval Warfare," and I wished to point out to naval officers, that whereas we could adopt, and ought to adopt and adhere to, Nelson's general principle, we could make a combination of which he could not have dreamt, because of the uncertainty of the elements and the uncertainty of his motive power. Now, on this point there was a reference made to Lord Charles Beresford's speech on the question of the manœuvres in the Channel last year. I recollect writing to one of the Lords of the Admiralty, when first the rules were published with reference to those manœuvres, and what I said to him was this:— "I do not know what the Admiralty intend to prove, but at all events I know what they will prove, which is that a slow fleet can do really nothing against a faster and more homogeneous one"; and that is exactly what took place. I venture to think, if that was a surprise to anybody, that it was not a surprise to our friend Sir Gerard Noel, because he also, I think, had foreseen that. Some people do not seem, even now, to appreciate the fact that that was the real question, and the whole question. Unless the officer in command of the more homogeneous fleet had lost his head altogether, and had not made use of the power placed at his disposal, he could not but succeed, and, *vice versâ*, the other man was practically bound to fail to make a good defence against a fleet which was so very much superior. Now I will pass to the question of submarines, and I can only say that I believe they will be proved to have their limitations. I cannot but recollect that when torpedo boats first came into general use, the French declared that they were perfect in all respects—that they could keep the sea in all weathers, and that they could always accompany battleships. We found later that they had distinct limitations. Now I should like to make one or two remarks on the question of engineers, and I cannot but regret that Mr. Laird Clowes has quoted Mr. Johnson. I must confess that Mr. Johnson is somewhat of a *bête noire* to me, because he has published several things in which he has assumed as facts things which are entirely in dispute, and he distinctly accused the admirals and the executive officers of being so hostile to the engineers, that they would rather let their ships be inefficient, and fail to maintain their speed, or fail to steam at all, than do any justice to that very ill-used class, according to his opinion. I confess I think he is rather out of court. With regard to the remarks made by Mr. Laird Clowes about the change which the Admiralty have recently made, I think it was a very wise change. I think we should, most of us, agree with President Roosevelt, who was at one time Secretary of the United States Navy. In a remarkable and, as I think, convincing report, he made use of this phrase: "In future, every naval officer must be a fighting engineer." Now, I believe, that is the goal which we must aim at, and I believe that the engineers will assimilate themselves more to the present naval officers, and the present naval officer more and more to the engineer. It is a matter of time. I do not approve of attempting to do wholesale what the Americans did, viz., amalgamate the two classes, but I do say that we ought to teach the naval officer much more scientifically. I agree with the author when he says that the scientific standard of our *personnel* should be very much advanced. Most decidedly it should be; but I would also remark, in connection with what has been said about the engineers, that we look upon them as an extremely valuable body of men. Many of them are personal friends of my own, and I have never had the slightest difficulty in dealing with the engineers. As regards their education at Plymouth—and I was some time at Plymouth, and had to look into the question of the Engineering College at Keyham—I must say, it struck me that their education was rather too technical, that is to say, they were almost too much mechanics. I speak under correction, and perhaps many engineers here, civil and others, will not agree with me, but it did seem to me that if we could have pushed their general education a little at the same time, it would have been an advantage. I recollect there was no question of teaching naval history at all. They spent six years in the College, but, except mathematics and their mechanical training, there was nothing else learnt; they said there was no time for anything else. That is open to question. At all events, if the engineers are to take their proper place, and to take as high a position as they aim at, and as I should like to see them reach, I think they ought to have a little more general education, just in the same way as we ought to have a distinctly higher scientific education amongst the present naval officers. Gentlemen, I have detained you long enough, and I only wish to remark in conclusion, that I regret that Mr. Laird Clowes has said—and I think it is almost a libel on the Admiralty to say so—that "the *personnel* has been assuming more and more the character of a penny-wise-pound-foolish makeshift." I do not know whether we are a penny wise and pound foolish makeshift, but I think that the present naval officers are endeavouring to learn all they can. Much may be faulty in the system under which they are educated; much might be added to it from a scientific point of view, but the steps which the Admiralty have recently taken, in giving over to the torpedo and gunnery officers those parts of the engineering work which are directly connected with the material with which they have to deal, are steps in the right direction, and I hope they will be carried further, and, if so, it will not be to the detriment of the engineer officers.

Professor J. H. Biles (Member of Council): My Lord and Gentlemen, in this very interesting paper there is one thing that we naval architects may congratulate ourselves upon, namely, that in the sweeping criticisms that are made, we have been pretty much left alone. Those Members of the Council who took upon themselves the responsibility of allowing this paper to come forward must, I think, have had in view the fact that we should look on pretty much as spectators in an arena in which the naval officer would be quite able to take care of himself, and protect himself from the writer's attacks. On the one side, we see that Mr. Laird Clowes charges the naval officer with not wanting so many things as he ought to want, and with not making use of some of the things that have been given him, even in spite of his not wanting them; while the naval architect, on the other hand, is very much disposed to charge the naval officer with wanting far too many things. The truth probably lies somewhere between the two. I think Sir John Hopkins did not, perhaps, quite understand Mr. Whiting in the very strong appeal that he made to naval officers for simplicity of idea. We all want simplicity of idea if we can possibly get it, but we want a great many ideas carried out, and it is not so much that each individual idea in itself becomes complicated, as that the naval officer wants so many ideas carried out. I think what Mr. Whiting had in view, and what naval architects generally have in view, is not so much objecting to the adoption of improved practical appliances, as the duplication, triplication, and quadruplication, of the many appliances that are put into a ship. When a naval constructor of Mr. Whiting's standing and experience comes to this Institution and deliberately says that there are a great many things—a long procession of them, I think he said—which might very well be displaced to the advantage of the armament, the subject should, I think, be seriously considered from that point of view. I am sure the Institution has listened with a great deal of pleasure to Mr. Whiting's remarks, and I think it is a great pity that we have not had the advantage of them in years gone by. At any rate, it is a consolation to know that although we have not Sir William White here, yet his absence has been marked by an exposition of the views of the naval constructors of the Admiralty, which I am sure was received with great pleasure. Now, on the subject of engineers, I think that a learned society is hardly the arena in which to discuss these personal matters. The naval officer is necessarily an engineer, more or less. An engineer who drives an engine and who is a trained scientific engineer is necessary in the ship, but it does not follow from this that the naval officer should not know as much engineering as possible, and, while it is not always possible to make the engineer and the naval officer the same kind of man, with the same kind of training, we shall probably get the best result the nearer they approach to each other. With respect to the range-finder (which I happen to know about from association with my colleague, Professor Barr), it seems to me that the position stands somewhat like this: the range-finder was demanded by the Navy, and the range-finder was produced. It fulfilled the conditions that were laid down. The Navy has taken it and has tried it, and has found out what can be done with it. They could not find out what could be done with the range-finder until it was developed, and they could not promise when it was developed that it would be perfectly and completely useful for the purpose for which it was intended. Now the Navy has, I suppose, at the present moment, found out the limitations of the range-finder, but to say that, because they have found out its limitations, therefore they have not encouraged the range-finder to its full extent, is not, I think, quite correct. If the range-finder had been as perfect an instrument as the Navy hoped, it would have probably been used more generally, and have been more highly appreciated than it is. But it still has to be perfected in the direction of finding the range when the ship is in rapid motion, and approaching a ship which is shifting very speedily. The ideal range-finder would be one which would be in the telescopic sight of a gun, and which would automatically change the sight as the distance of the enemy altered. That is the ideal range-finder. It may be developed some day, but it is not so yet, and, until that day, I think we can fairly say that the Navy has treated the range-finder in an encouraging and proper spirit. One looks at the general question of encouraging inventors rather from the point of view of what is done in other countries. Now, I think Mr. Laird Clowes is very close to the mark when he says that we do not encourage inventors. Our tendency is rather to look upon an inventor as some one who should not be given too much encouragement, but, on the other hand, I think that the inventor is rather apt to look upon a person who wants something as an individual to whom he should take an idea and leave that individual to work it out. The American practice is rather that those who are responsible are perfectly willing to talk over and discuss freely the ideas of an inventor, but they will not accept his ideas alone, nor will the inventor generally put forward his invention in the form of a completed article until it has been thoroughly and completely tried. That is the American practice: first invent something, then have it made and completely tried, and then put the thing forward to the person who wants to use it. Inventors in this country have not advanced as far as they have in America in the direction of producing complicated articles, and I think that is perhaps the reason why those who have to use the inventions have not encouraged inventors so much as they do, at any rate, in the United States. I think we owe a debt of gratitude to Mr. Laird Clowes for giving us an opportunity of hearing some very interesting criticisms from naval officers upon the criticisms that he has showered upon them.

Mr. F. Edwards (Member): My Lord and Gentlemen, as an inventor, I wish to state most clearly that I do not agree with Mr. Laird Clowes' statement about the attitude of the Admiralty towards inventors. My experience is directly opposed to it, in connection with inventions.

Mr. W. Laird Clowes (Associate): My Lord and Gentlemen, I can imagine the Prophet Daniel bearing up against the lions, for he was a man of exceptional qualifications. I am not a Prophet Daniel, and I am in a den of naval officers and naval architects; but, owing to the unlooked for kindness with which they are treating me, I still hope to survive the ordeal! I was exceedingly interested in Mr. Whiting's remarks; and most cordially do I agree with him that simplicity is most desirable. But efficiency is still more so; and my contention is, that if any invention will add to the efficiency of a result, one ought not to neglect it merely because one feels naturally prone to cling to simplicity. Sir John Hopkins alluded to the proposed employment of the gun against the submarine. That is not my suggestion, it is the suggestion of Commander Kimball, of the United States Navy, who, I believe, has had more experience than anyone else alive with submarine boats. He seems to think that before any submarine which has been to the surface can disappear even a few inches below the water, a torpedo-boat destroyer or a small cruiser ought in many cases to be able to hit her with a gun, if it be properly mounted so as to reach her when she is close under the side of the ship. With regard to the black powder ignition charge for heavy guns, I may have been misled by some facts which appeared in a recent number of the Proceedings of the United States Naval Institute, and which, perhaps, apply only to the ignition charges which are found to be necessary with the smokeless powder used in the United States' Navy. If so, I am very sorry. But, surely, it ought not to be necessary to employ smoky powder at all. Sir Edmund Fremantle, whom I shall always remember as one of the earliest and most consistent advocates of the advantages of speed, has alluded to the fact that we are behind other nations. Unfortunately there is no question of that. My state of health has not allowed me in recent years to spend part of every summer, as I once did, on board some British man-of-war at sea; but, of course, I have been on board men-of-war in port. Now I remember several years ago visiting an American man-of-war and seeing electrical shot hoists which had been fitted in her. Last summer, at Portsmouth, I went on board the *Cressy*. I was taken to the *Cressy* because I was told that I ought to see the very latest thing. The very latest thing in the *Cressy* was an electrical shot hoist very similar to what I had seen in an American vessel several years earlier, and which had given, I believe, the greatest satisfaction. Yet the *Cressy* was the first British ship to be commissioned with electrical shot hoists. If I may trust my memory, she must have been five or six years at least behind the American competitor. Admiral FitzGerald declares that we—that is, the country and the Navy—do appreciate the strategical value of speed——

Admiral FITZGERALD: I said the Navy; I am not responsible for the country.

Mr. LAIRD CLOWES: Well, let us speak of the people who are responsible for the Navy. I say the proof of the pudding is in the eating. I say that if the Navy really appreciated the strategical value of speed, the Navy would have the fastest sea-going ships in the world; and I add that the Navy has not got them. I say, further, that the Navy would be careful to have the fastest mercantile (auxiliary) cruisers in the world; and again, that the Navy has not got them: out of twenty-two existing sea-going merchant vessels which could be used as cruisers in warfare, and which have a speed of twenty knots or over, only nine belong to this country. That is a fact which indicates that the Admiralty does not appreciate the strategical value of speed. It ought to know that it will need such vessels in war time, yet it does not assure to the country the possession of vessels which shall have the heels of foreign ships. With regard to the range-finder, it is doubly clear, after what Sir John Hopkins has said, that such an instrument is very necessary in our Navy, seeing that we have not got an unlimited number of Captain Scotts. Nevertheless, as a matter of fact, the range-finder cannot yet be regarded as a regular Service instrument. Its use is not yet, I mean, part of the ordinary gunnery curriculum of every naval officer and seaman. It has only been experimentally adopted by us, although it has been used in other navies for years. It may be a very complicated thing; it may be a thing very liable to get out of order in action; but why not use it as long as you can? If it should get out of order in action, you have still your old-fashioned methods to fall back upon. As to Admiral FitzGerald's point that the idea of the old-fashioned blockade has not been entertained by his naval friends for the past twenty-five years, all I can say is that a few years ago I participated more than once in British manœuvres, of which a close—not merely a watching—blockade was a very conspicuous feature. I seem to remember Admiral Fitz-

Gerald himself, when a captain, not so very long ago, taking part during the annual manœuvres in another blockade. No, the *idea* of the blockade is not dead in the Navy. But, in spite of all that Admiral Fitz-Gerald has said about my paper, I am greatly obliged to him for having with such vigour led the discussion, which has been extremely interesting. I know perfectly well how unfitted I am to read here any paper except one which shall have the effect of irritating better qualified persons into an expression of their views. Well, I believe that this time I have made a good many people speak; and, as they have spoken very kindly of me, I have no cause for anything but satisfaction.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G.): Gentlemen, Mr. Laird Clowes appears to think he has rubbed people up the wrong way, and so started a good discussion. I am sure we all agree that he has written a paper which has produced a very interesting discussion, and I am sure also that you will all join with me in giving him a hearty vote of thanks for the trouble he has taken.

DISTORTION IN BOILERS DUE TO OVERHEATING.

By C. E. Stromeyer, Esq., Chief Engineer of the Manchester Steam Users' Association, Member of Council.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 21, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

Overheating of boiler plates through shortness of water, scale, and grease is one of the dangers which engineers have constantly to bear in mind, but it does not appear that the subject has been exhaustively examined, and I therefore hope that inquiries may be encouraged by a review of the natures and difficulties of the problems.

It is well known that boilers which may work satisfactorily under ordinary conditions will give trouble when the fires are forced. It is, therefore, desirable first of all to ascertain what is the meaning of forced fires, and how these affect furnace plates with and without scale. A fair rate of combustion under natural draught is 20 lbs. per square foot of grate per hour, while 40 lbs. is a fairly high rate with forced draught. There is, however, no limit to the air pressure, and the combustion can, if necessary, be increased to 80 lbs. and probably to 100 lbs. per square foot of grate per hour, but under such conditions the products of combustion which leave the bed of fuel will doubtless consist almost exclusively of carbonic oxide and nitrogen with little or no carbonic acid, and unless much air is then admitted above the bars, combustion will be incomplete and the temperature low.

Professor A. Ledebur* has shown that, if charcoal be burnt at a low temperature, much carbonic acid and little carbonic oxide are produced, while much oxygen remains unconsumed, whereas at a temperature of about 2,000° Fahr., the product of combustion is practically pure carbonic oxide, all the oxygen being consumed. If, under these latter conditions, we estimate the temperature of the resultant mixture of nitrogen and carbonic oxide, we find that it should be about 2,400° Fahr. Now, by introducing just sufficient air into this mixture to re-ignite it and effect perfect combustion, the temperature rises to about 4,200° Fahr., say 2,200° C. This is the maximum

* "Stahl und Eisen," 1882, Vol. II., p. 356.

possible temperature with a coke or anthracite fire, and then only if the second addition of air is made before the gases are at all cooled by radiation or by contact with the boiler plates. This condition can only be attained in ordinary boilers with thin fires, the excess of air passing through the larger interstices of the fuel, through local heaps of ashes, or through air spaces which are sometimes formed between the furnace plate and the outer fire-bars. In brick combustion chambers or furnaces, the flame itself doubtless attains a maximum temperature, whereas in ordinary boiler furnaces the maximum temperature is sometimes found in the bed of fuel, and sometimes in the flame just above the fuel, much depending on the amount of air admitted and the position of the inlet.

In Belleville boilers with economisers, the furnace temperature is kept low by incomplete combustion and re-ignition of the gases just under the economisers. The thickness of the fires is so regulated that much carbonic oxide is generated, which, after being cooled amongst the lower tubes, mixes with an additional air supply and burns again. Stoking has to be done most carefully, otherwise all the heat is generated in the lower furnace and the lower tubes get overheated.

In a forge fire the temperature is probably a maximum, and it is under that condition that Mr. Yarrow, according to my estimate,* evaporated 142 lbs. of water per square foot per hour. This evaporation, according to Miss Bryant's experiments,† corresponded to a temperature of about 3,870° Fahr., the formula being $(\Delta t)^2 = H . 90{,}910$ where Δt is the difference of temperature between the plate and the fire, while H is the heat, in evaporative units, transmitted per square foot per hour. In Mr. Yarrow's experiment, the temperature of the fire side of the plate would be about 140° Fahr. in excess of that of the water, so that the flame would be about 4,220° Fahr. In view of the agreement of these estimated furnace temperatures, we may assume that 4,000° Fahr. is a maximum.

Mr. Yarrow's evaporation was estimated by me from the acquired curvature of this tube plate, the corresponding radius ρ, being 550 in., which was due to the fire side of the plate being warmer and expanding more than the water side. If now we consider the case of a cylindrical furnace exposed over its upper half to a fire of the maximum intensity, and if we suppose this furnace to be slit along its lines of fire-bars, then, on account of the increased temperature on the fire side, the upper half would uncurl. If its original diameter was, say, 40 in. $= 2\,r$, then its acquired diameter would be 2 R, and we have—

$$\frac{1}{R} = \frac{1}{r} - \frac{1}{\rho} = \frac{1}{20} - \frac{1}{550} = \frac{1}{20 \cdot 8},$$

showing that the furnace width has increased to 41·6 in., and the furnace crown

* "Marine Boiler Management and Construction," p. 127.

† Ibid., p. 120.

has been lowered by 0·8 in. In addition to this change of form in a circumferential direction, the furnace crown would also tend to bulge down longitudinally, but being prevented by its cylindrical shape, the circumferential distortion would be increased about 30 per cent., making the horizontal diameter 42·1 in., and the vertical one 37·9 in. If now we re-connect this furnace top to its bottom, the deformations will be halved, and the furnace would have a horizontal diameter of 41·05 in. and a vertical one of 38·95 in., the difference of the two diameters being 2·1 inch. The furnace ends or strengthening rings would unquestionably have some influence in reducing this deformation, but the above consideration makes it clear that a furnace which was originally somewhat flat would be weaker than one whose vertical diameter was originally slightly large.

The increase of the mean temperature of the upper half of the furnace tends, as was shown experimentally by the late Mr. Lavington Fletcher * to bend the furnace as a whole, and this action would naturally add to the flattening effect just mentioned.

The next point to be considered is, what are the stresses set up in a water tube if surrounded by a fire of maximum intensity? Clearly these stresses are the same as would be set up in a plate which is being bulged to a radius of 550 in., and that stress is—

$$S_1 = \frac{E \cdot t}{2 \cdot \rho} \cdot \frac{\mu}{\mu - 1}$$

where

E = Modulus of elasticity = say 31,000,000 lbs. per square inch.
t = Thickness of plate in inches.
ρ = Radius of curvature in inches.
$1 : \mu$ = Co-efficient of cross contraction.

S_1 is, therefore, 40,000 lbs. per square inch $\times\ t$ = 17·8 tons per square inch $\times\ t$, which means that the fire side of the tube of one-inch metal is subjected to compression stresses of 17·8 tons per square inch, both longitudinally and circumferentially, while the water side is subjected to equally intense tension stresses. These stresses are due to the temperature gradient in the metal, and depend on the heat transmission and on the thickness, but not on the diameter, of the tube. The thickness, however, depends on the diameter D, and the pressure P; if then we fix on a limit of, say, 10,000 lbs. per square inch as being the maximum permissible strength for a water-tube exposed internally to a pressure P, of 200 lbs., and externally to a fire of maximum intensity, we easily find the limiting diameter as follows :—

10,000 lbs. per sq. in. = $S = S_1 + S_2$ = sum of stresses.

* "Report on a Series of Red-Hot Furnace Crown Experiments," Manchester Steam Users' Association, 1889, page 19.

$$S_1 = 40{,}000 \, . \, t \, ; \qquad S_2 = \frac{P \, . \, D}{2 \, . \, t} = \frac{200 \, . \, D}{2 \, . \, t} = 100 \, . \, \frac{D}{t}.$$

$$10{,}000 = 40{,}000 \, . \, t + 100 \, . \, \frac{D}{t}.$$

$$D = 100 \, . \, t - 400 \, . \, t^2.$$

Differentiating, we get—

$$\frac{d\,D}{d\,t} = 0 = 100 - 800\,t.$$

From this it follows that D is a maximum and equal to $6\frac{1}{4}$ in., when $t = \frac{1}{8}$ in. In other words, for lesser thicknesses than $\frac{1}{8}$ in., the tube diameter has to be made smaller than $6\frac{1}{4}$ in. on account of the stresses due to internal pressure; for greater thicknesses, the diameter has to be reduced on account of the stresses caused by the temperature gradient in the metal.

Of course, this deduction is only true for those cases where the furnace temperature is a maximum, or equal to or higher than that found in a forge fire, but to judge by the molten surfaces of the firebricks in some dry back and water-tube furnaces, these temperatures are not infrequent with these types of boilers when worked with strong draught. This also explains why gas fires whose maximum temperatures are about 2,000° Fahr. do not appear to give much trouble under water-tube boilers.

In the above-mentioned cases the heating surfaces were supposed to be clean. By covering their water side with scale or grease, the conditions are altered considerably. Roughly speaking, scale of $\frac{1}{10}$ in. thickness offers as much resistance to the passage of heat as does a film of grease $\frac{1}{100}$ in. thick, or a plate of steel 10 in. thick. Now, in order to transmit as much heat through one square foot of boiler plate as will evaporate 20 lbs. of water per hour, the temperature gradient must be 40° Fahr. per inch of thickness; in a layer of scale the gradient must be one hundred times greater, or 4,000° Fahr. difference on either side of a thickness of 1 in., or 400° Fahr. difference for a thickness of $\frac{1}{10}$ in. If, then, we have a clean furnace plate $\frac{1}{2}$ in. thick, transmitting 20 evaporative units of heat per hour, it will be 20° Fahr. hotter on the fire side than close to the water, and its mean temperature will be 10° Fahr. greater than that of the water and other parts of the boiler. If the intensity of the fire be increased so as to raise the heat transmission to 140 evaporative units, then the mean temperature of the clean furnace plate will be 70° Fahr. higher than that of the water and the rest of the boiler. But 10° Fahr. and 70° Fahr. produce respectively an expansion of 1: 18,000 and 1 : 2,600, which, if entirely prevented by rigidity of the boiler structure, will produce stresses of 1,700 and 12,000 lbs. per square inch respectively.

If now we add $\frac{1}{10}$ in. of scale on the water side, and increase the intensity of the fires so as to maintain the above-mentioned evaporations, there will still be the same temperature gradients in the plates, but, added to these, there are a hundred times steeper ones in the scale, and the water temperature being fixed, we arrive at a mean plate temperature of 410° Fahr. with 20 lbs. evaporation, and of 2,870° Fahr. excess, with 140 lbs. evaporation. If the fires are not intensified, and that is of course impossible where the temperature is a maximum, the reduced difference of temperature between plate and fire will in the first case lower the evaporation, and, therefore, also the plate temperatures, by about 15 per cent.; in the second case the reduction will be nearly 30 per cent., so that the plates, when covered with scale, instead of expanding forty-one times as much as when clean, will only expand about thirty times as much, or, say 1 : 530 and 1 : 87 of their dimension. This means that if the plates were held rigidly, stresses of nearly 60,000 and 350,000 lbs. per square inch respectively would be produced. Of course, long before these deformations are attained, something in the boiler must give way. If it is the furnaces that are covered with scale, probably the front end plate will get pushed out, and the motion, if often repeated, will ultimately crack the furnace connections: this is frequently the case with land boilers. This expansion also sometimes leads to furnace corrugations. If it is water tubes that are coated with scale, they will either bend, a most frequent occurrence, and perhaps tilt their ends out of their holes, or they may force their ends further through the tube holes. To guard against these troubles, even water tube boilers should be designed so as to be fairly elastic, especially as the changes of temperature of the heated tubes may be very sudden.

It seems to be vaguely believed that if a plate be coated with scale or grease, these substances, being bad conductors of heat, should exert at least some beneficial influence by preventing sudden changes of temperature; but it must be remembered that scale isolates the plates from the water, and that they have now to change their temperatures in harmony with the fire fluctuations, each opening of the door and each shovelful of coals causing serious cooling and contraction, while every time that the fire is freshened up, the plate temperatures rise again.

Grease, as already mentioned, is a ten times more effective non-conductor of heat than is scale, and all the above estimates on scale of $\frac{1}{10}$ in. thickness would be equally true of a film of grease $\frac{1}{100}$ in. thick.

If now we deal with hard scale, as we have at first dealt with furnace plates, and take into account the change of form produced by transmission of heat, we naturally arrive at the result that hard scale should have a tendency to detach itself from the plate on which it has grown, not only because it expands at a different ratio to steel, but also because of the bending action due to transmission of heat; but the greatest influence is, perhaps, the partial drying when the scale has grown so hot as to superheat its moisture.

A very instructive case came under my notice.* During a spell of exceptionally dry weather, the ordinary water supply for a boiler gave out, and water from a discarded well had to be used. It appears to have been of a brackish nature, salt and scale rapidly accumulated in the boiler, until the furnace bulged in two places, one bulge tearing. The steam pressure was less than 40 lbs., showing that the furnace plates must have been very hot. When I visited the boiler, the scale over the unrent bulge was as yet undisturbed; it retained its original cylindrical or arched shape (see Figs. 1 and 3, Plate XXII.), while the furnace plate was bulged as shown. The hollow space must have been filled with superheated steam, which, being a bad conductor of heat, must have allowed the plate to become red-hot. As long as the scale was in contact with the plate, the temperature of the latter would be sensibly the same as that of the scale, and as there would be a considerable temperature gradient in the scale, the plate may easily have been a few hundred degrees hotter than the water in the boiler. As soon as separation took place between the plate and the scale, the moisture, which would be oozing through the scale, would become superheated, and the plate would then find itself sandwiched in between this superheated steam and the furnace gases, and would acquire their mean temperature, which might easily be equal to a red heat.

In this case the scale was hard and not very thick, about ½ in., and although the conditions for separating it from the plate must have been more than usually favourable, it is certain that in all boilers even thin flakes of scale are frequently separating from their heating surfaces, and therefore the process noticed by me in this case may have been the cause of many other bulges. In water-tube boilers we experience a very similar action. (See Fig. 2, Plate XXII.) If hard scale be allowed to form in a tube, say under easy working conditions, and should the boiler then be worked to its utmost capacity, the iron or steel of the tube will grow hotter than previously, the scale in contact with the plate will be partially calcined, which would facilitate separation, and, as the tube expands more than before, a thin space filled with superheated steam will be formed. The metal is now removed from all cooling action, and swells until it bursts, or until the scale cracks and admits water, which boils up suddenly and of course carries the scale away. Although scale is a bad conductor of heat, overheating of scale-covered plates does not take place as often as might be expected, which may perhaps be accounted for by its porosity and by a natural circulation of water or steam which takes place in scale. Under favourable conditions small craters are formed, which are evidently the orifices through which the steam and water which have penetrated to the plate through the scale are discharged.

* Board of Trade Report, No. 1,130.

An instance of the porosity of scale was shown by the case already mentioned (Report, No. 1,130). When I examined this furnace, it was in the condition shown in Figs. 3 and 4, Plate XXII., the scale projecting very considerably over the part of the plate which was bent down, and it seemed marvellous that the boiler should have practically emptied itself of water without carrying away the projecting scale. On my return to the office, I found that our inspector, who had been at the boiler before me, had brought with him some scale which had fallen through the rent in the furnace on to the matting which had been placed inside. The area of this scale was about equal to the area of the hole in the scale as seen by me, so that the choice now lies between two somewhat improbable alternatives; either there was only a small opening in the scale, and yet the steam and water rushed through this opening without breaking the surrounding unsupported scale, or else there was no hole in the scale, and the steam and water found their way through the porous scale, which, being arched and not broken, was fairly strong. The steam pressure was believed to have been fairly low, and certainly very little damage was done to surrounding property.

I have made estimates, which may prove of interest, as to the time required to overheat a plate when exposed to the action of a fire. Miss Bryant's most valuable experiments* show that the one surface of a heated plate is only a few degrees hotter than the water with which it is in contact, but the fire must be much hotter than the plate, so as to transmit an equal quantity of heat. If now we replace the water on one side by a gas, like steam, the plate will rapidly acquire a temperature, which would be about the mean of the furnace and steam temperatures, the latter, of course, rising if the supply of steam be limited. The rate at which the plate temperature rises, at least at first, can be roughly estimated on the assumption that it receives heat from the fire, but loses hardly any to the steam. The specific heat of iron being about 0·11, an evaporative unit of heat (= 966 B.T.U.) entering an iron plate 1 ft. square and 1 in. thick (weight 40 lbs.) would raise its temperature by 220° Fahr. We know that at a temperature of about 600° Fahr., the elastic limit of mild steel is reduced to about one-half of what it is when cold, while at about 1,200° Fahr. the elastic limit is reduced to about one-eighth. If we consider a furnace plate ½ in. thick exposed to a moderate heat, such as will evaporate 20 lbs. of water per square foot per hour, then if the water level be lowered so as to expose the plate, its strength will be reduced one-half in the same time that 1⅜ evaporative units would have been transmitted. The time required to do this would be 8¼ minutes. Therefore, after exposing a plate to the above-mentioned heat conditions for 8¼ minutes, bulging might be expected at any time, and in 17 minutes such a furnace plate would only retain one-eighth of its original strength,

* Proceedings of the Institution of Civil Engineers, 1897, Vol. CXXXII., p. 274.

and would most certainly have given way, unless the original factor of safety for compression was greater than 8.

If, as in water tubes, the metal is only $\frac{1}{8}$ in. thick, such temperatures as 600° and 1,200° Fahr. will be attained respectively in one-quarter of the periods just mentioned, or, say, in two or four minutes; but as such tubes have a factor of safety of from thirty to forty, one need not expect bulging before two minutes have elapsed, and even after four minutes, rupture need not necessarily take place. The fact that bulges are not infrequent, shows that such tubes have to stand much more than is generally believed.

When the fire is as fierce as it can possibly be made, the periods necessary for heating the plates or tubes are reduced to about one-seventh of the above, and have to be counted in seconds. Short though these periods are, they are not negligible, and have to be remembered when dealing with such bulges and bursts.

Reference has already been made to grease in boilers, but the subject is known to be one of so much difficulty, and even mystery, that it requires special attention. There can be no doubt that the introduction of grease into boilers may cause furnaces to bulge or to groove, water-tubes to bulge and burst, and stay and smoke-tube ends to leak, yet when these injured parts are examined, grease may be said to be conspicuous by its absence, although present in other parts of the boiler.

It also seems to have been noticed that grease has a more marked effect in clean boilers than in boilers covered with scale, and it is quite certain that grease is far more injurious with forced than with natural draught. There are also many experiences which show that collapses due to grease are very gradual, extending sometimes over months, and, not unnaturally, not only one but several furnaces then collapse together. Thus, in my own experience, all nine furnaces in three boilers of one steamer bulged during a prolonged trial trip, and had all to be renewed, while in a recent water-tube case two to six tubes were generally found bulged at a time, renewals being necessary at the rate of 114 in fourteen months in five boilers.

The thermal conductivity of paraffin is 0·41, of gutta percha 1·4, and of plaster of Paris 4·8, measured in British thermal units. The latter substance has the same chemical composition as hard boiler scale, whereas grease is of the nature of the two hydro-carbons named, and may roughly be stated to be a ten times better non-conductor of heat than hard scale. An estimate has already been given of the mean temperature of a furnace plate covered with a scale $\frac{1}{10}$ in. thick, which was found, including the steam temperature, to be about 700° Fahr. when there was a reasonably strong fire, evaporating 20 lbs. of water per square foot of heating surface. With a heat as intense as that in a smith's fire, the mean temperature was shown to be about

3,200° Fahr. But furnace scale of $\frac{1}{16}$ in. in thickness is not a very serious matter, and no practical engineer would point to it as being the cause of a furnace collapse. This means that the heat transmission through the plates of our furnaces is generally less than 20 evaporative units per square foot per hour, otherwise, with slightly thicker scale, the mean plate temperature would be above 1,000° Fahr.

If for the scale we substitute a grease of ten times greater non-conducting power, but of about one-hundredth of an inch in thickness, then we may expect such a plate temperature not to be exceeded; yet furnaces in a greasy boiler collapse when there is practically no grease on them. I, at least, who have seen many bulged and collapsed furnaces in greasy boilers, although I have then found much grease on the stays, on the shells, especially near the water-line, and on the under side of the furnaces, have never yet found even a film of grease on the furnaces where bulged, nor even on the furnace parts above the fire-bars; and we are therefore driven to the conclusion that grease, when in a boiler, undergoes a chemical change whereby it becomes perhaps a thousand times worse conductor of heat than it was; or, and this is more probably correct, the action of grease when preventing the passage of heat is of a totally different nature to the action of scale.

One explanation offered is that a thin film of grease, if heated as in a boiler, rises off the plate in the form of blisters, which blisters are filled either with oil vapour or superheated steam, and act as did the detached scale patches referred to in Fig. 1, Plate XXII., and that the skins of these blisters are then floated away. Mr. Austin, of Lloyd's Register,* even goes so far as to express the opinion that furnace collapses are due to large bubbles even when no grease is present.

Another view is that furnace collapses are due to a compound substance consisting of finely divided scale and iron rust and grease, which, having accidentally acquired the same specific gravity as water, floats about in the boiler and adheres to the hotter plates, and then falls off again when collapse has taken place.

Others again maintain that some collapses at least are due to scum, consisting of grease and light sediment, which floats on the surface, but settles on the furnaces when the boilers are emptied and then re-filled without cleaning. I have been through thousands of boilers, many of them unscaled, and though I have looked for the two substances just mentioned, I have never seen them.

It is also maintained that collapses are due to floury deposit, said to be oxide of magnesia, which is precipitated from sea water in the presence of caustic soda. have not come across this substance in marine boilers.

* Transactions of the Institution of Shipbuilders and Engineers in Scotland, Vol. XLI., p. 257.

One serious objection to all these theories is the one already stated, that collapses due to grease are gradual, and can under favourable conditions be watched growing for months; they therefore differ essentially from collapses due to scale or to shortness of water, for in these two cases, when the plate has become so much overheated as to commence bulging, its change of shape constitutes such a material structural weakening that complete collapse takes place.

I regret to say that I cannot bring forward a satisfactory explanation of collapses in greasy boilers, but, as the action is a very gradual one, it would seem as if the amount of overheating were not necessarily very great, otherwise the furnace would be too weak and would collapse before it could recover itself; for the same reason the force producing the deformation cannot well be a permanent nor a widespread one, and would appear to be local, and more intense than the steam pressure. These limitations of choice seem to point to bulges in greasy boilers as being due to some sort of hammering action connected with ebullition.

The view that hammering goes on in boilers when worked very hard, will probably be new to many marine engineers, but in France, the home of water-tube boilers, the matter is treated as being of common knowledge. Thus M. C. H. Bellens* mentions blows as occurring in water-tube boilers. It should also be remembered that in the Belleville boiler water-hammering is made use of to assist, or, as some maintain, to create circulation, a non-return valve being fitted at the foot of each element of tubes.

The Board of Trade Commissioners, in their summing-up of the previously mentioned case,† remark that it was stated in evidence that rumbling noises were heard in these boilers, and, on making inquiries, I learnt that not only rumblings, but loud reports as of water in steam pipes were frequent, and that these sounds came, not from the steam pipes, but from the boilers. Sharp blows could also be distinctly felt when standing on the boiler drums. It may, therefore, be taken for granted that, under certain circumstances, severe hammering does occur in a boiler, but what those circumstances are is not so easy to say.

Two explanations suggest themselves, and I can at present offer no other; these are, retarded ebullition and water-hammer action.

In order to grasp at least the possibility of retarded ebullition, one has to consider the question of ebullition generally. If one had never seen water boiling, and if one were asked to describe what one would expect to occur if water of a boiling temperature

* "La Mécanique à l'Exposition de 1900," pp. 24 and 25.

† Board of Trade Report, No. 1,130.

were placed in contact with a hotter plate, one would be tempted to say that, at first, innumerable small bubbles would form all over the surface, like dampness on a window pane; that these bubbles, like the water drops on the glass, would increase in size, and would join together, and when an individual bubble had acquired a sufficiently large size, it would break away and rise to the surface, the space thus left bare being at once re-covered with minute bubbles. It is, however, well known that water does not boil like this; on the contrary, when only little heat is applied to the plate, one may have to wait for long periods between the appearances of bubbles, and, when they do show, they grow so suddenly, and tear themselves away so quickly, that the process cannot be followed by eye. It will, however, be noticed that, on its upward journey, each bubble increases in size, especially at first. This cannot be due to the decreasing head of water over the bubble, for it would have to rise 30 ft. before its diameter is increased by 30 per cent. The increased size cannot be due to the inertia of the water; for, assuming that the bubble is formed with such rapidity as to impart an appreciable velocity to the mass of water above it, and that, in coming to rest, the water exerts a suction on the bubble and thus increases its size, then, clearly, after the first increase, the bubble would decrease again. This does not seem to be the case; besides, a suction which would double the volume of the bubble would have to be about 7 lbs. per square inch, and, for however short a period this reduced pressure might exist, it would cause the whole body of the nearly boiling water to burst into steam.

Only one further explanation suggests itself, and that is, retarded ebullition; which means, that the water is superheated. If water be heated in a glass beaker, one will notice thread-like transparent swirls moving about near the heating surface, which are due clearly to differences of temperature, density, and optical properties of hot and cold water. These swirls are, of course, less marked when the boiling point is reached, and when the whole mass of water is of a nearly uniform temperature; but they re-appear during the period of formation of steam bubbles, and can be seen if relatively very little heat be applied. These new swirls are, to my mind, superheated water, whose density, or at least whose optical property, differs markedly from non-superheated water.

Suppose the bottom of a beaker full of hot water, to be covered with a layer of superheated water, and suppose that, for some reason or other, this superheated water gives up its steam at a particular point, perhaps because of the glass being thinner or rougher there; then suddenly a fairly large bubble would be formed, which, while it rises, creates an upward current, and further bubbles are likely to appear at the same spot. Now a bubble, one-tenth of an inch in diameter, which is fairly large under these conditions of boiling, could be formed by the partial evaporation of a volume of water the size of a shilling, and superheated to only $\frac{1}{50}°$ Fahr.

Rough surfaces, as is well known, prevent, or at least reduce, retarded ebullition; grease is believed to increase it, though I confess that my experiments have only partially confirmed this view; on the other hand, certain chemical precipitates, especially those which quickly settle, are such well-known retarders of ebullition, that chemists are most careful, when boiling waters containing them, to constantly stir them. Quite recently this matter was brought to my notice by what might almost be called an explosion in the chemical laboratory of the Manchester Steam Users' Association. Our chemist was boiling some water containing a precipitate of oxalate of lime, when suddenly the contents of the beaker were shot upwards to the ceiling through a height of 8 ft. above the table, the beaker being shattered. It seems that this behaviour of the precipitate is a fairly certain one, and almost equally sudden ebullition occurs on boiling after precipitating albumen by the addition of permanganate of potash and caustic potash.

The question of interest in this case was the amount of superheat and the pressure exerted. Assuming that the beaker did not break until after the fluid had left it, then the pressure of steam, which was suddenly generated from the layer of supposed superheated water, would be equal to the depth of fluid multiplied by the ratio of the height of 8 ft., to which it was lifted, to the height through which it moved while still in the beaker. This estimated pressure height was found to be 4 ft. to 6 ft., corresponding to two or three pounds per square inch, and to from 6° to 9° Fahr. superheat. These estimates are quite independent of the volume of superheated water, except that, to produce the upheaval, the layer of superheated water must have been deeper than one six-hundredth of an inch, a not unreasonable supposition. The same result could not have been obtained with a lower degree of superheat, no matter how large the volume of superheated water may have been.

One difficulty stands in the way of unreservedly accepting this explanation, and that is, that the sudden ebullition shattered the beaker. Now it is difficult to believe that this breakage occurred after the water had left the beaker, that is, after the pressure had been relieved, and therefore the propulsion must have taken place during the very short interval of time that it took to break the glass bottom. This interval can be estimated with the help of the explanation which I recently gave as to how to calculate the pressure exerted in steam pipes subjected to water-hammer action.* The time and pressure would depend on the elasticity of the fluid and of the glass, and also on the respective velocities of sound in either material. My lowest estimate of the pressure exerted in this accident is 400 to 600 lbs. per square inch, and the time during which it acted must have been so short that only the top layer of the water in the beaker could have been shot up. But

* Manchester Literary and Philosophical Society Memoirs, 1901, Vol. XLVI., No. 3, p. 15.

(400 + 15) lbs. per square inch corresponds to about 240° Fahr. superheat, which it is quite unreasonable to expect to have existed, not only in this experiment but in any conceivable case. Therefore, I see no alternative at present but to accept the provisional assumption, which has already been characterised as somewhat improbable, that the beaker held together as long, or nearly as long, as the water was still in it, but then cracked, and that the explosion phenomenon was due to only a few degrees of superheating. But as soon as we refuse to admit the possibility of intense, though local superheating, we are face to face with the difficulty that mild superheating cannot account for the powerful concussions heard in water-tube boilers, and then there seems to be no alternative theory other than that of water-hammer action.

Before, however, going into this matter, I would like to suggest that it is not the grease coating on the heating surfaces which causes retarded ebullition, or more correctly the very violent local ebullition, but that this is more probably due to some fine precipitate produced by the interaction of the grease dissolved—not suspended—in the water on the iron or on the rust of the iron of the heating surfaces, in such a manner as to form a precipitate which, similar to oxide of manganese, oxalate of lime, and others, causes retarded ebullition. Should this view be found to be correct, and should it be possible to discover the conditions for producing this injurious deposit, then possibly we may also find means to remove those conditions and be relieved of the danger of the action of grease.

Let us assume the inside of a water-tube to be coated with a substance which retards ebullition, also that the water circulation is sufficiently sluggish to allow of a reasonable degree of superheat, say 10° Fahr., and suppose that in a 4 in. pipe we have a film of this superheated water measuring $\frac{1}{32}$ in. in thickness, and covering 1 ft. of the length of the pipe, or say 1 sq. ft. in area. Then, when ebullition does take place, we have suddenly a volume of about 10 cub. in. of steam. Ten degrees superheat corresponds to a pressure of about 17 lbs. extra pressure—the ordinary boiler pressure being assumed at 150 lbs.—and although this pressure must rapidly diminish while the bubble is being formed, and will finally become negative, yet we see that there is here sufficient force to drive the columns of water in the tube in both directions. Their motions will not be arrested, as one would on first thoughts imagine, when the volume exceeds 10 cub. in., for a local diminution of pressure of only 5 lbs. below the general pressure of 150 lbs. would lead to the generation of a volume of steam equal to one-half the volume of the water which was subjected to this diminution. Experiments on glass-tubed models seem to support this view, for the volume of the bubbles sometimes formed in such tubes is greater than simple retarded ebullition could be made to explain, and there seems to be no serious objection to suppose that, for short intervals of time, steam spaces, one or more feet in length, are formed in water tubes of boilers. That is certainly a belief held by many engineers, even without examining the question from the above points of view.

Assuming, then, that a fairly large volume of steam has been suddenly brought into existence in one of the lower tubes of a water-tube boiler, there will most certainly be considerable commotion in the two columns of water which have just been torn asunder. Now the water in the lower part of the lowest tube will occasionally contain an excess of comparatively cold feed water, which excess will be the less, the more rapid the circulation, but which excess will also have had the less time to be warmed up if the circulation is rapid. On account of this excess of relatively cold feed, which for argument's sake will be assumed to amount to one-twentieth of the water in the water-tube boiler, a lowering of the temperature of the steam in the bubble amounting to 15° Fahr. would at once be effected. The large steam bubble would at once commence to condense against the colder water column, and a further lowering of pressure would take place, resulting both in a closing up of the steam space formed, and in a tearing of the upper or hotter column of water; the intervening plug of water would be accelerated towards the cooler column under a difference of pressure of 25 lbs. per square inch, corresponding to the above-mentioned reduction of temperature of 15° Fahr. If we assume the steam bubble to be, say, 24 in. long, and the plug of water to be 12 in. long, then the latter will strike against the lower and cooler column with a velocity of 91 ft. per second, and, according to the method of estimating the intensity of the resultant blow as previously mentioned (see foot-note, page 204), the local pressure would be—

$$P = E \cdot \frac{91}{W} = 160{,}000 \cdot \frac{91}{3{,}200} = 4{,}550 \text{ lbs.} = 2 \text{ tons per square inch.}$$

Here—

$E = 160{,}000$ lbs. is the elastic modulus of water at 150 lbs. pressure with a temperature of 358° Fahr.

$W = 3{,}200$ ft. per sec., the velocity with which sound in pressure waves travels in water when at a temperature of 358° Fahr.

It is not suggested that this pressure of 2 tons is the actual pressure which may frequently be exerted in a water-tube in which retarded ebullition may have led to two columns of water meeting with a shock, but the assumptions on which this calculation is based are not so unreasonable as to suggest that 2 tons per square inch is an impossible pressure. These assumptions are :—10° Fahr. superheat, a sudden steam bubble expanding to one about 24 in. long, and that there is sometimes 5 per cent. of cold feed water in the downtake.

Supposing that only 1° Fahr. of superheat were possible, even then the resulting pressure would be 450 lbs. per square inch, which might suffice to produce slight bulging, more particularly as the tube is supposed to be exposed to an intense fire, frequently denuded of water, and it may be prevented from parting with its heat by the same deposit which is supposed to produce or accentuate retarded ebullition. The same pressure would be produced if either the bubble were only $2\frac{1}{4}$ inches long, or only $\frac{1}{2}$ per cent. of cold feed water were in the downtake.

In spite of these arguments, they cannot, I fear, be accepted as fully justified, for they leave much to be desired when applied to the collapsing of furnaces of greasy boilers. With such collapses, the strongest argument in favour of the above explanation is that they occur gradually, and are therefore likely to be caused, not by uniform overheating and weakening of the plates, which would result in rapid and complete collapses, but by local excesses of pressure, which, according to the above views, would be severe blows, but inaudible, or at least unnoticed, when the engines are working hard, the only time when the fires would be at their maximum intensity.

One very serious objection to this view is, that the water in a marine or Lancashire boiler (more especially above the fire-bar line of the side furnace, when there are three in a boiler) is hardly likely to be colder than the rest of the water, so that the suction which has to cause the water-hammer cannot occur, or must be very slight, unless (and this possibility should not be lost sight of) there is a chance of the cold feed-water descending on to the furnaces. It may be urged that the superheated water is most probably lighter than the colder water, and would tend to move upwards either on the furnace plate or separate from it. This objection would be met if it could be shown that superheated water bursts into steam if moved, or that it tends to cling to the spot where it was formed. It can be urged that a furnace is a very much weaker structure than a water tube, more particularly as regards resisting local blows, and that, on account of the much greater thickness of its metal, very severe stresses already exist in it, due to transmission of an assumed large quantity of heat; and, lastly, the pressure produced when a bubble disappears by the closing in of surrounding water is very much greater than when two plugs of water meet in a tube. In the one case the blow is, as it were, met at a point, the centre of a hemisphere, whereas in a tube it is met by a surface, so that very much smaller causes than those which were assumed in the case of the water tube, either as regards superheat, volume, or cooling, may produce local pressures which are as intense, if not intenser, than those just dealt with. This means that numerous very minute but exceedingly intense blows may constantly be occurring on the furnace plates of boilers fed with greasy water. A soap bubble bursts practically without noise, yet one can hear water boiling, therefore this noise must have its seat in the water, or, possibly, on the heating surface; and, further, when water commences to boil, and when, therefore, the bubbles formed re-condense at once on reaching a somewhat cooler layer of water, we either hear the well-known singing noise, indicating that certainly a few hundred bubbles are formed and collapse every second, or else we hear sharper knocks indicating the formation and annihilation of fewer but larger bubbles.

It will be seen that although the bulging of tubes and furnace-plates in greasy boilers may possibly be explained as being due to innumerable blows of considerable intensity, it cannot be said that this view is free from serious objection, and I would not only welcome any other suggestion, but I am sure that the Institution could desire nothing better than that experiments dealing with this subject might be made. Years ago, marine engineers had gained a point when it was discovered that grease in boilers was one cause of collapses, but, as grease still finds its way into boilers, yet seems to disappear from those plates which it is believed to have injured, we ought, I think, not to be satisfied with a vague statement, but should search after the true nature of the action of grease.

DISCUSSION.

Mr. J. I. THORNYCROFT, LL.D., F.R.S. (Vice-President): The author, I think, has gone very minutely into many particulars that interest us greatly, and he has not only given them in general terms, but he has given us the results of his calculations. He has shown us that it is really possible to calculate where it might seem very difficult to do so. The one particular thing I wish to speak of, however, is the deformation of a furnace due to oil or some other substance resisting the passage of heat to the water. I wish to show that there is another theory which commends itself to my mind, and which seems to explain how the furnace may perhaps be deformed very gradually, without the water-hammer action which the author has described, or other forces than those due to simple heating and cooling of a flue in which you get intense heat. Take, for example, a round, ordinary furnace or flue in which there is a fire: what I wish to show is, that it is possible, or at least highly probable, that the heat gradients at different parts of the flue are essentially different, the steepest gradients being near the side of the fire. This arises from two causes: firstly, the hot gases or flames impinge almost directly on the surface of the flue, and, giving up heat, pass obliquely along the flue and upwards, thus protecting the crown or upper portion of the furnace from the direct impact of the flame from below; the other cause being, that the sides of the flue are nearer to the incandescent fuel, so that the radiant heat on this part is also greater. The first effect of this steep gradient of temperature at each side of the flue, before the metal has been so much heated as to cause a stress beyond the limit of elasticity of the material under the circumstances, is to temporarily cause a change of form in which the vertical dimensions are increased more than the horizontal (the latter may, perhaps, be even reduced), but if the heating be sufficiently intense and continued, the plate at the sides will become so hot through part of its thickness as to lose much of its resistance to crushing, and so become permanently shortened, when cold. This effect applies equally to a circumferential or a longitudinal section. The immediate effect of this change is to reduce the stress in the other parts of the plate, allowing the cooler parts to take the form approaching that in which the plate is not in stress, so that the flue flattens and increases in width beyond its proper circular section; any oil or non-conducting coating which divides the metal from the surrounding water, will, of course, much facilitate this action. In order to carry this action further, it is necessary to assume that conditions can arise in which the stress now induced in the plate forming the crown of the furnace (when cold) can be at least partially removed, and that would appear to be possible when the crown of the furnace is more thickly coated, and the fire has less intensity. By repeating the cycle described, it seems possible to explain the very gradual change of form of tubes placed in marine boilers, where oil finds access in sufficient quantity to allow the overheating described.

Mr. MACFARLANE GRAY (Member of Council): My Lord and Gentlemen, without a careful study and a quiet perusal of this paper, no one can express a correct opinion regarding it. Mr. Stromeyer is a careful observer and a clever reasoner, and we may congratulate ourselves on having the chief engineer of the Manchester Steam Users' Association as a member of our Institution, giving us the benefit of his wide experience. The paper he has just read is one of great importance, and some of its new thoughts will be often referred to in future papers in this Institution.

Mr. A. F. YARROW (Vice-President): I am very pleased to have the opportunity of congratulating Mr. Stromeyer upon the value that must be attached by the Institution to his paper. I should like to ask one or two questions. On page 202, the following passage occurs:—"Sharp blows could also be distinctly felt when standing on the boiler drums. It may, therefore, be taken for granted that under certain circumstances severe hammering does occur in a boiler, but what those circumstances are is not so easy to say." I have had to do with many water-tube boilers, but have never found any hammering such as that referred to. Then again, on page 205, the author states:—"We are face to face with the difficulty that mild superheating cannot account for the powerful concussions heard in water-tube boilers, and then there seems to be no alternative theory than that of water-hammer action." I have not had any experience of these powerful concussions. I do not know if Mr. Thornycroft, or anybody else has, but we do not find that is so in the boilers we have to do with, and I should very much like to know whether, in the boilers Mr. Stromeyer refers to, the tubes are nearly vertical or nearly horizontal.

Mr. STROMEYER: They are Babcock boilers.

Mr. YARROW: That is, very nearly horizontal?

Mr. STROMEYER: Yes.

Mr. YARROW: Perhaps the inclination of the tube may have something to do with this concussion. Now, on page 203, he says: "It will, however, be noticed that, on its upward journey"—that is, a bubble of steam—"each bubble increases in size, especially at first. This cannot be due to the decreasing head of water over the bubble, for it would have to rise 30 feet before its diameter is increased by 30 per cent." I can quite understand that if it were a bubble of air, it would only increase proportionately to the reduced pressure on itself, and it would have to travel up some distance before there would be any very considerable increase of size; but with a bubble of steam, surrounded by boiling water ready to fly into steam, the case is entirely different, because, the boiling point of water at the lower portion of a column of water being higher than at the upper part, steam is formed as the water rises, without the absorption of additional heat. The bubble of steam not only expands as it travels up, but it also increases in size, because of fresh steam joining it from heated water flying into steam owing to the reduced head and lower boiling point. If you have a nearly horizontal tube, I can quite imagine that there is a large body of heated water, travelling along comparatively slowly, at a temperature which is not sufficient to produce steam with that particular head of water, but which, when travelling upwards slowly through a tortuous course, will very suddenly fly into steam, and give rise to the powerful concussions referred to in the paper. I should, therefore, not be surprised to find, in boilers where the tubes are nearly horizontal, that the conditions are such as to produce powerful concussions which do not exist when the circulation is very direct.

Mr. JOHN DEWRANCE (Visitor): My Lord and Gentlemen, Mr. Stromeyer has given us an excellent explanation of the way in which a tube is overheated. But he says that he does not exactly see (or so I understand him) why the grease that is found to be the cause of trouble in a boiler should be absent from that part which it has injured. Now, it seems to me that Mr. Stromeyer himself has supplied the materials for the explanation. When scale is formed inside a tube, we have, as he fully explained, the superheated steam upon the outside pressing the metal, which has become red hot, into a deformed state. The tube then suddenly ruptures, and, as a rule, the scale is broken by the force of the steam inside. I never saw a case, such as Mr. Stromeyer was fortunate enough to see, where the scale remained. We find that part of the tube which has been deformed perfectly free from scale, and it has sometimes that black oxide which proves the way in which it has been deformed. For exactly the same reason, the grease is absent from that part which is injured, because it is not the grease alone which causes the injury, but a combination of grease and scale. It may be extremely thin, but a certain proportion of grease in the scale will prevent the water from wetting the metal, and will very likely produce, with an eighth of an inch of scale, an amount of distortion of the tube which would not be produced by perhaps three-quarters of an inch of porous scale. That seems to me, therefore, to be the explanation of why, when the tube is distorted, there is no longer any grease on the part which is injured. I think, my Lord, it is extremely interesting that this fact should have been brought out by Mr. Stromeyer in such a public way, as I believe it will explain a great many of the injuries to tubes which have previously been held to be mysterious.

Mr. F. EDWARDS (Member): My Lord and Gentlemen, I wish to thank Mr. Stromeyer for this paper, and to say that I consider it to be one of very great value. I was recently called in to look at some boilers in which the repairs, and consequential losses, amounted to about £8,000, through the furnaces coming down. There is a question which I would like to ask, as I do not see it mentioned in the paper, and which, I think, would be well worth the author's attention. Why is it that furnaces frequently come down a certain distance and then stop there? If they are jacked up, they come down again to the same place, and remain in that position. This frequently occurs. My impression—and I have been trying to ascertain why it is—is this: the furnace, in the first instance, will just take a certain number of bars, the latter being a tight fit; then, when these bars are heated, they expand and set up a considerable stress, pushing the sides outwards until the latter have made room for the bars; to permit of this, the crown has to come down a certain amount, and ultimately takes a permanent set sufficient to allow room for the bars in their heated state. When room has been given, the stress which caused the furnace to come down is removed, and this accounts for its not coming down any further. Then, if the bars are taken out and the furnace is jacked up to its original form, and the same number of bars are put in again, the same heavy stress in the horizontal direction occurs, and brings the furnace down again. I believe that, in the old days, the way in which the furnaces were set up was to put in a vertical bar about six inches square, and make a fire round it, and the expansion of the bar set the furnace up—instead of using the jack. This appears to me to be very similar to what takes place horizontally when the bars are put in tight. If Mr. Stromeyer can give us any information on that point, I, for one, shall be very grateful.

Mr. C. E. Ellis (Associate Member of Council): My Lord and Gentlemen, it is very rarely that a non-technical member like myself can interpose in the discussion on a paper of so scientific a nature as Mr. Stromeyer's, but it happens to have fallen to my lot to be able to present to the Institution what I think is a very remarkable illustration of the dangers which one has to expect in marine boilers, and I cannot help thinking that it will be of interest to the Institution if I give some particulars of what I am referring to. I am, amongst other things, considerably interested in the manufacture of steel, and partly in the manufacture of marine boiler furnaces, and in one particular case we supplied a number of furnaces for a vessel. Your Lordship will understand that I do not wish to mention any names in connection with this matter, but there was no doubt that, by some mistake, something went wrong with the water supply of the boilers, and there was a general collapse of nearly all the wing furnaces of those boilers. Now the furnaces were what are well known as Purves's type, and I am sure you will understand me when I say that I absolutely disclaim any suggestion that what the Purves furnace stood in this particular instance would not have been withstood by any other furnace in the market. The furnace in question was 3 ft. 7 in. in diameter, and when the water supply failed, the crowns of all the wing furnaces were, of course, terribly exposed to heat; the pressure came on and they collapsed. Now they collapsed, in one instance to my knowledge, to a distance of 2 ft. down; in other words, instead of a circumference of 3 ft. which they started with, the lowest part of the crown of the furnace came 2 ft. down from where it had been. The effect upon the steel was such that the part between the ribs, which was 9 in. in length, was stretched in one case as much as 2 in., so as to bring it to 11 in., and the thickened rib was flattened out in certainly a most extraordinary way. The photographs and sketch (Plates XXII.A and XXII.B) of those furnaces give an idea of the extent of the damage, and they go to prove the absolute necessity of taking all the precautions which Mr. Stromeyer has pointed out in his paper to be advisable in dealing with this matter. I ought to add that there was no tear whatever in the furnace after collapse; not a rivet was broken, showing how well the boiler had been made, and there was not even a rivet hole damaged by the collapse.

Mr. C. W. J. Bearblock, R.N. (Visitor): My Lord and Gentlemen, I should like to ask the author to give us some further explanation on a couple of paragraphs in his paper. On page 194, there is a statement about the Belleville boiler and the use of the economiser. The last sentence in the second paragraph states that the stoking must be done very carefully, otherwise all the heat is generated in the lower furnace and the tubes get overheated. This view of the functions of the gas mixer and space below the economiser is interesting, but in the working of the boiler, complete (or approximately complete) combustion is often effected in the space immediately above the fire. Besides, the boiler has certainly been subjected without injury to bad stoking without use of the gas mixer below the economiser. In another paragraph, on page 202, the author says:—"It should also be remembered that in the Belleville boiler water-hammering is made use of to assist, or, as some maintain, to create circulation, a non-return valve being fitted at the foot of each element of tubes." That is apparently an error, as they are not fitted at the bottom of the generator elements to which this remark appears to apply. On page 196, the author deduces equations involving the thicknesses and diameters of tubes. He deals with a thickness of $\frac{1}{8}$th of an inch, and I should like to see his results for tubes of $\frac{3}{16}$ in. and $\frac{1}{4}$ in., if he would interpret the equations for those thicknesses.

Mr. J. I. Thornycroft, LL.D., F.R.S. (Vice-President): I only rise, my Lord, to answer the question that Mr. Yarrow asked with regard to the hammering or noise in the tubes of boilers with which I have been connected. I must say, like Mr. Yarrow, that I know nothing of hammering in the tubes. If the water passes through with sufficient rapidity, I believe hammering does not take place, but in tubes in which the water passes through rather slowly, and if it happens by any chance that feed water of a comparatively low temperature gets jerked into the large bubbles he was describing, it will cause sudden collapse of the steam forming the bubbles, and hammering, such as is often heard in buildings which are heated by steam, when the pipes are not so thoroughly drained as to keep the cooler water from being in the same pipes as the hot steam. But, so far as I know, that has never taken place with any boiler with which I have been connected.

Mr. C. E. STROMEYER (Member of Council): My Lord and Gentlemen, I think it will be best if I begin at the end, and reply first to the last speaker but one. As regards the formula on page 196, I cannot give the results for any other thicknesses. I show there that $\frac{1}{8}$th of an inch is the minimum thickness. If the tube is made thicker, it is weaker because of the heat strain; if made thinner, it is weaker because of the boiler pressure; $\frac{1}{8}$th of an inch is the least and also the best thickness for the particular conditions. Clack valves are, as far as I know, always fitted in Belleville boilers. All the drawings that I have seen and all the papers that I have read on the matter show or mention these valves, and they seem to be very essential to the working of these boilers. It may be that some of the modern boilers are worked without them, but I think it is the usual practice to fit clack valves. As regards stoking, I think it is well known that the Belleville boiler has to be stoked most carefully. The boiler is a very efficient one if properly worked, that is, if the stokers are well trained and the fires are kept exactly at the right thickness. I have not, personally, supervised any experiments, but I understand that if the fires are not worked properly, and if, for instance, too much coal is placed on the grates, it will be consumed, but there will be no additional steam production. There is one particular condition of fire which gives the best results, and, with the economiser in use, I understand that result is an excellent one. If such boilers are put in charge of men who are not skilled in working these very large furnaces, the results are, I believe, very uneconomical. That may account for the varying returns that are obtained in the Navy, some boilers being economically worked and others not. With regard to Mr. Edwards' inquiries, I think the explanation why bulges recur in the same place lies in the straining to which the material has been subjected, while bulging and while being pressed back. A slight bulge is a small flat, and therefore shorter than the segment from which it was formed. When it is pressed out again, the metal must be in tension, and if, through scale or other causes, it gets overheated again, it comes down again.

Mr. EDWARDS: Why does it stop there?

Mr. STROMEYER: That is one of the puzzles of the action of grease in boilers to which I have tried to draw attention. The experience of the Manchester Steam Users' Association, both with land boilers and marine boilers, is that the bulging process may go on slowly for years. Our inspectors report the shapes of all furnaces, and, if irregular, they are especially watched.

Mr. EDWARDS: I meant the whole furnace.

Mr. STROMEYER: Occasionally there are bulges, and occasionally the whole furnace gets flat, but we watch both. Perhaps in the second year, we may find that the bulge has increased an inch or so, and we take steps accordingly. We warn the owners to watch the bulges and not to put so much grease into the boiler. The bulging action sometimes goes on and sometimes stops. I think we have altogether about 200 boilers whose furnaces have deformed or are deforming, and we keep careful records of them. It is one of the difficulties of the problem that the action is so very slow. I have been trying to suggest explanations, and I hope, as Mr. Macfarlane Gray says, that they will lead to further inquiries, but I cannot yet give a definite explanation of it. I think this also answers one of Mr. Thornycroft's inquiries, except that Mr. Thornycroft's own explanation, if gone into more in detail, may, I believe, be found to be the correct one, and I would not be at all jealous of a better suggestion being brought forward. If I understand him correctly, the irregularity—the uncurling—which I mentioned before may be local, and therefore would not weaken the furnace as a whole. A hot patch here and there may start bulges which gradually increase. I think if that theory were worked out, it might be made to explain also bulged water tubes. The difficulty with my suggestion is that, although it seems to fit the water tube troubles, it does not answer so well with the furnace troubles, and I think Mr. Thornycroft's explanation may get over that difficulty, because it may perhaps fit both cases. Unless an explanation does fit both cases, I do not think that it should be accepted as quite satisfactory. I have, I believe, not been quite explicit as regards the absence of grease, but it is not only the bulged parts of furnaces that have no grease on them, it is the whole furnace top that has no grease. Although, in touching these parts, one's hands get black, this is due to a dark dust, and very little of that. There is certainly no grease; but below the fire bars, the furnaces are sticky with thick grease, and in the case I referred to, of nine furnaces coming down in a ship during a trial trip, I knew, before I entered the boilers, that there was grease in them, for the overalls of the men who came out were perfectly greasy. Nearly the whole of the inside of the boiler was greasy, but not the furnace crowns, which had bulged. The difficulty is, how to explain why these furnaces collapsed. One knows that the prime cause was grease, but, notwithstanding this, the collapsed parts were not greasy. That is one of the mysteries we have to deal with. My suggestion is that, whether the cause of the collapse is water-hammer or overheating, these causes are due to some chemical precipitate caused by the action of the grease on the iron. As regards the hammering action which Mr. Yarrow and Mr. Thornycroft have not heard, I fancy very few people at sea would hear it, because, so far as I understand, the boilers in which hammering was noticed were worked practically at their utmost capacity, but then they were separated a long distance from the engine. There were thick walls in between, and you could be in the boiler house without hearing the engines working at all, and, standing on those boilers, the engine builder told me he could feel distinct blows under his feet, and the stokers told me that, standing in the stokehold, they could hear blows in the boilers. When a torpedo-boat boiler is being worked at its very highest power, the engine is also doing the same thing, and makes itself so much felt and heard that concussions would not be noticed. I do not mean to say that I am convinced that the concussions which were heard were the causes of bulges and rents, but, seeing that they are supposed to assist circulation in the Belleville boilers, I believe that they do at least occur, and my paper is an attempt to ascertain whether they are really serious factors.

Mr. YARROW: I was referring to the trials of water-tube boilers ashore. The engine question does not come in at all.

Mr. STROMEYER: Then you did not hear blows?

Mr. YARROW: No.

Mr. STROMEYER: I simply mention that the blows have been heard, but what causes them, I do not feel sure about.

Mr. A. SPYER (Member): May I ask the author if he could tell us, in that case, where the feed was introduced?

Mr. STROMEYER: It would be in the top of the boiler, the usual system adopted by Messrs. Babcock and Wilcox.

Mr. Spyer: I have never heard this hammering in a water-tube boiler except in one case, and there it was distinctly traced to the position at which the feed water was introduced.

Mr. Stromeyer: That is very interesting, I will mention it to the people who have these boilers, and perhaps that will get them over the difficulty. I felt some hesitation in bringing forward the suggestion that the superheated water is the true explanation of the ebullition, not for fear of its being criticised, but because the question of superheating was at one time thought to be the cause of all explosions, with which view I have not the least sympathy. I mean that, formerly, whenever there was a boiler explosion, it was at once urged that it was due to retarded ebullition, which is but another name for superheated water. That theory, I hope, has been absolutely knocked on the head, especially by my predecessor, the late Mr. Lavington Fletcher. But I do not believe that superheating is entirely inconsistent with water boiling, and I am putting forward that theory to assist in explaining how collapses might occur. Mr. Yarrow adopts the explanation given, I think, by Professor Tyndall as to the inner working of geysers: he assumes that before an outburst takes place, the water temperature will be higher at the bottom than at the top, at the rate, it seems to me, of about $1\frac{1}{2}^{\circ}$ for every foot of depth. In a beaker 4 inches deep, half full of water and just on the point of boiling, the surface temperature would be 212° Fahr. against 212·25° Fahr. at the bottom. A bubble, on rising, should increase in size, but not to the extent which actually seems to be the case. Now, although I agree with Mr. Yarrow that there is an increase of temperature with the depth, I do not think that this can fully account for the increase in the size of the bubbles, nor for the repeated appearance of bubbles at one point. When once a bubble has formed at one point, a vertical current is set up there. Then the next time that a bubble is formed at the same point, it will find to hand a column of superheated water all the way up to the surface, and I think it is the amount of superheated water which bursts into steam, which causes those bubbles to increase in size. I think, therefore, that Mr. Yarrow and I agree about ebullition, except that he attaches importance to the necessary temperature gradient, while I believe in superheated water. This is only a slightly different way of looking at the question. I have, my Lord and Gentlemen, to thank you very much for the way you have discussed my paper, and I hope it will lead to further results.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G.): From the way in which Mr. Stromeyer's paper has been received, I am sure you will all join with me in giving a hearty vote of thanks to him.

ON THE CORROSION OF CONDENSER TUBES AND SEA-WATER CONDUCTORS.

By Professor Ernst Cohen, of Amsterdam.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 21, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

I have accepted with much pleasure your kind invitation to give an account of my researches on the corrosion of condenser tubes and copper sea-water conductors on board steamers, a trouble that people in your country also have had to struggle with for many years. At the very outset, however, I must point out that experiments to test practically the efficacy of the means I have recommended to remedy this evil are now being made, and that the object of the inquiries cannot be considered to have been reached until the results prove that the means recommended are a final solution of the problem.

The phenomena which occasion so much trouble are familiar to all of you. Whilst on the inside the brass condenser tubes of steam boilers are in contact with sea water, they are, on the outside, in contact with steam and water which has been formed by condensation. In the walls of these tubes sometimes, even after a short time, holes appear, which necessitate renewals; moreover, the metal parts of pumps and copper sea-water conductors are rapidly and strongly corroded.

As the reports on these phenomena, which were given to me by companies interested in the matter, contained a great deal of contradictory information, I decided to get acquainted with the phenomena by personal observation. Besides the conflicting conclusions which had been drawn with respect to the causes of the corrosion, in a great many cases even the mere facts did not tally. This is to be ascribed to the fact that the information was founded upon accidental observations; systematic researches have apparently not hitherto been made.

(1) The Action of Sea Water on Brass and Copper.

In researches like those which will be described here, it is of great importance to choose the circumstances of the experiments as similar as possible to those in which the phenomena actually appear, or even to take more favourable circumstances. Now, in reality, sea water runs through copper (or brass) tubes which have the temperature of the air or that of the engine-room. In the case of the covering of the condenser, sea water runs through brass tubes which assume a temperature of 38° C. Of course the temperature of the water will also become about 38° C. For this reason my experiments were made at this temperature. In order to keep this temperature up for a long time I used a thermostat, a reservoir of about 80 litres, filled with water which was constantly stirred by a small propeller, kept in motion by a hot air motor. (See Fig. 1, Plate XXIII.) A thermo-regulator regulated the gas flame so that the fluctuations of temperature remained within 0·03 of a degree. In this thermostat the flasks and apparatus were suspended.

(2) Does Sea Water Act upon Pure Copper?

In connection with the general opinion that the corrosion of copper by sea water should be ascribed to impurities, I endeavoured to ascertain how far sea water acts upon chemically pure copper. The sea water used for these experiments was taken from the North Sea; in a few cases I took it from the Mediterranean. As I expected, there was no difference in the action of either. Chemically pure copper, which had been specially prepared for this purpose with the greatest care, was plunged in sea water, and by a special arrangement (see Fig. 2, Plate XXIII.) atmospheric air was allowed to bubble through the water. Varying the circumstances in different ways, as has been described in my Dutch paper on this subject (*de Ingenieur*, 16 March, 1901), I was able to observe the following facts:—Chemically pure copper is only corroded by sea water if atmospheric air can co-operate in the action. If the carbonic acid of the air is removed, the action does not take place.

(3) The Action of Sea Water on Commercial Copper.

Experiments analogous to those described in §.(2) were made with—

(*a*) Electrolytic copper.
(*b*) Sheet copper.
(*c*) Hammered copper.
(*d*) Cast copper.

The copper is only corroded if sea water, air, and carbonic acid act upon it simultaneously.

(4) The Action of Sea Water on Brass Condenser Tubes.

I repeated the experiments with brass in sheets and brass condenser tubes, after having made an analysis of these materials. The composition was :—

Copper	... ...	66·60 per cent.
Zinc	... ...	33·40 „

Here also I found that the corrosion by sea water only takes place if the air and carbonic acid act simultaneously upon it. When the action goes on, both the copper and zinc are dissolved by the surrounding fluid. I will spare you the chemical details; let me only say that a short calculation shows that the quantities of carbonic acid which are present in sea water, according to the best experiments in this direction, are sufficient to cause the phenomena which have been mentioned.

(5) The Action of Sea Water on Tin-plated Condenser Tubes.

In order to prevent the corrosion of condenser tubes by sea water, some of these tubes have been tin-plated. To what extent has this plating a protective action? As there appeared to be no actual researches about the action of sea water on tin, I also carried out some experiments in this direction which gave the following result: Tin is corroded if it is in contact with sea water and atmospheric air.

I also determined by chemical analysis the thickness of the tin layer which covers tin-plated tubes of different manufacture, and found it to be only $\frac{1}{50}$ of a millimetre. I also observed the fact that the tin layer, even of perfectly new tubes, is not always quite intact. In some places circular spots are found where the brass is bare. The origin of these spots is probably connected with the method of manufacturing these tubes, which are tin-plated by plunging them into melted tin and then withdrawing them quickly. If the inside was not quite clean, it may easily happen that the tin will not adhere uniformly everywhere.

(6) The Action of Sea Water on Different Materials.

Special experiments showed that oxide of copper and nickel resist even a prolonged action of sea water and atmospheric air, so that if it were possible to cover condenser tubes with these materials they would resist destruction. Aluminium bronzes, however, which have been recommended by several authors, are attacked very strongly in a short time.

It seems strange that the corrosion is often quite of a local nature, while the metal near it seems to remain quite sound. In order to get an insight into these processes, a well-polished brass plate was set up in a large glass tube, which contained dilute nitric acid. This liquid was constantly stirred by a glass propeller, worked by a hot-air motor. Although this solution remained quite homogeneous, yet the plate was not corroded evenly, but holes were formed with wedge-shaped edges. If we take into consideration that homogeneity is a very relative notion, and that we call a plate homogeneous when viewed with the naked eye, which, in reality, is not homogeneous at all, this result is not so surprising.

(7) Protection of Copper and Brass Tubes by Zinc or Iron.

Owing to the results obtained I was led back to the researches of your famous countryman, Sir Humphry Davy, who described, in the years 1824 and 1825, a great many experiments about the protection of ships, by copper sheathing, against the corrosive action of sea water. His researches, of which more recent technical literature has apparently taken no notice, led him to the result that the contact of copper sheathing with zinc or iron entirely prevented the evil.

If a piece of copper or brass is exposed to the combined action of sea water and atmospheric air, corrosion occurs. If, however, under otherwise similar circumstances, the copper (or brass) is brought into contact with a very small piece of zinc or iron, the action of the sea water is entirely suppressed. It is not necessary at all that the copper (or brass) should be in direct contact with the zinc or iron. In order to prove this, take two glass cylinders A and B (Fig. 3, Plate XXIII.) filled with sea water, and connected by a U-shaped tube C equally filled with the liquid. Into A we put a piece of copper or brass, and into B a very small piece of iron or zinc, and connect the copper and zinc by a copper wire. The brass remains quite bright, whilst the iron (or zinc) is dissolved.

In order to note the influence of the electrical resistance of this combination upon the protective action of the zinc, I used the apparatus shown in Fig. 4. A and B are two flasks, which are connected by the glass tube S S S, filled up with sea water; d_1 and d_2 are copper wires, hermetically fixed by means of sealing wax into glass tubes in the rubber stoppers of the flasks. By attaching and starting an air pump, air flows into the flask A through the tube l and passes through Z Z. A small piece of brass (or copper) P_1, is now fixed to wire d_1 and in the same manner a small piece of zinc in the flask B to the copper wire d_2. When air is going through A, the corrosion of P_1 occurs. If, however, we bring m_1 and m_2 into contact beforehand by a metal wire, the corrosion is entirely suppressed, the brass (or copper) remains bright, whilst the zinc in B is corroded. If the electrical resistance of the water in the tube S S S is too large, the protective action of the zinc disappears.

By varying the length of the tube S S S we can find at what resistance the corrosion of the copper re-commences.

If the brass plate P_1 corresponds with the condenser tubes in the condenser on board a ship, and P_2 with a piece of zinc which is suspended in the sea, then S S S corresponds with the tubes connecting the condenser tubes with the sea, viz., the tubes supplying the sea water which is used for cooling. If now the zinc plate and the condenser tube are connected by a copper wire, we have a system like that which has been described in Fig. 4 (Plate XXIII.) where corrosion cannot occur.

If the resistance between the two electrodes of Fig. 4 becomes too large, the protective action of the zinc disappears. This seems to point to the fact that the intensity of the electric current in the system is playing a part. By increasing the intensity of the electric current by means of an auxiliary battery, the protective action of the zinc is re-established.

If m_1 is brought into contact with the positive pole of the battery, corrosion takes place immediately, as chlorine is evolved from the salt of the sea water; if, however, the brass plate is in contact with a piece of zinc, this will be corroded, but the brass remains quite bright. This fact proves that, if, in the condenser tubes, there should happen to be any electrolysis, &c. (which of course would produce chlorine), the presence of zinc would, in this case, also prevent the corrosion of the tubes.

Direct electric currents, produced by leakages from dynamos on board, even if they are very small, may give rise to corrosion of the tubes in the manner described. The phenomenon will only be produced by this cause if the condenser tube is connected to the positive pole of the dynamo.

Diegel* concludes from his researches into this subject that zinc and iron will act as protectors, and this agrees with Davy's experiments and those described in this paper.

As a result of the researches mentioned here, I should recommend the following:—

(8) Remedies.

(*a*) A thick coating of tin on copper or brass tubes.

(*b*) A coating of oxide of copper. It is not certain, however, that this remedy is capable of practical application.

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* *Marine Rundschau*, November, 1898: Jahrgang 9; 1485-1550.

(*c*) A coating of nickel on the inside of the tubes. It would, however, be preferable to use nickel tubes instead of brass or copper.

(*d*) An electrical connection between the copper or brass tubes and zinc or iron plates which are suspended in the sea.

(*e*) A continuous electrical current, accurately controlled, going through the condenser tubes, or through those parts of the ship which have to be protected against corrosion.

(*f*) Insulation of the tubes from all parts of the electrical installation in which currents flow.

DISCUSSION.

Mr. A. F. YARROW (Vice-President): My Lord and Gentlemen, I have nothing to say, except to thank Professor Cohen for his kindness in coming here to read this paper. I have some records of experiments which may be of interest in connection with this paper and which are therefore appended in Tables I. and II.

EXPERIMENTS WITH CONDENSER TUBES AND BOILER TUBES BY MESSRS. YARROW & CO.

The objects of these experiments were :—

(1) To try the effect of tinning and nickel plating on brass condenser tubes.

(2) To try the action of fresh and salt water on condenser tubes, under various conditions of temperature, pressure, or gas in solution.

The specimens were all of the same size, and were experimented upon at the same time.

All the specimens were weighed every ten days, with the exception of the sealed specimens, which were weighed after 275 days.

In the case of the sealed specimens, the hydrogen or air was only present in a small quantity between the stopper and the surface of the water; the object aimed at was to seal the specimens in the boiling water with as little air as possible.

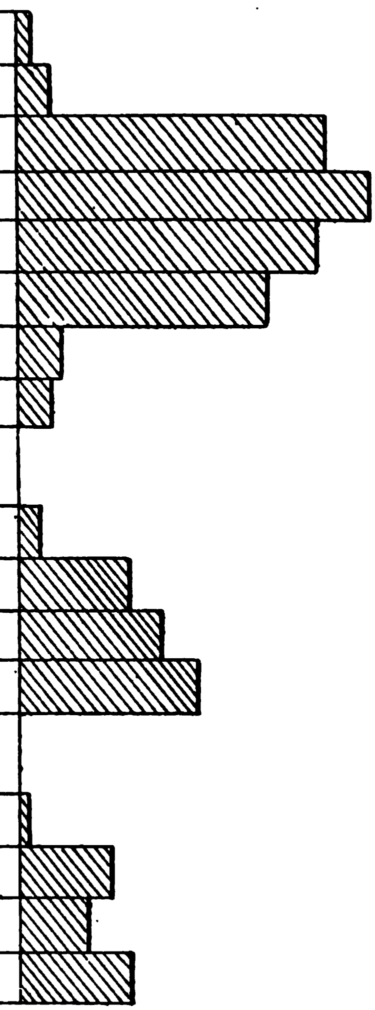
In one of the nickel-plated specimens, the nickel plating at one end was defective and this tube pitted badly at this end.

With the exception of this nickel-plated specimen, the corrosion was more or less uniform over the surface; in some cases it made the tube look striated longitudinally.

TABLE I.

EXPERIMENTS WITH CONDENSER TUBES BY MESSRS. YARROW & Co.

Kind of Tube.	How Tested.	Loss per sq. inch per ten days in milligrammes.	Relative amount of loss per ten days shown shaded.
Ordinary brass Condenser Tube.	Sealed with boiled sea water and hydrogen ...	·06	
	Sealed with boiled sea water and air	·14	
	Sea water open to air	1·54	
	Sea water with air bubbling through	1·82	
	Sea water at 100° F. and open to air	1·54	
	Sea water at 10 lbs. pressure ... (This pressure was not constant.)	1·27	
	Plain water at 10 lbs. pressure ... (This pressure was not constant.)	·20	
	Distilled water at 10 lbs. pressure (This pressure was not constant.)	·16	
Tinned brass Condenser Tube.	Sealed with boiled sea water and air	·11	
	Sea water open to air	·54	
	Sea water with air bubbling through	·66	
	Sea water at 100° F. and open to air	·88	
Nickel-plated brass Condenser Tube.	Sealed with boiled sea water and hydrogen ...	·04	
	Sea water open to air	·45*	
	Sea water with air bubbling through	·36	
	Sea water at 100° F. and open to air	·55	



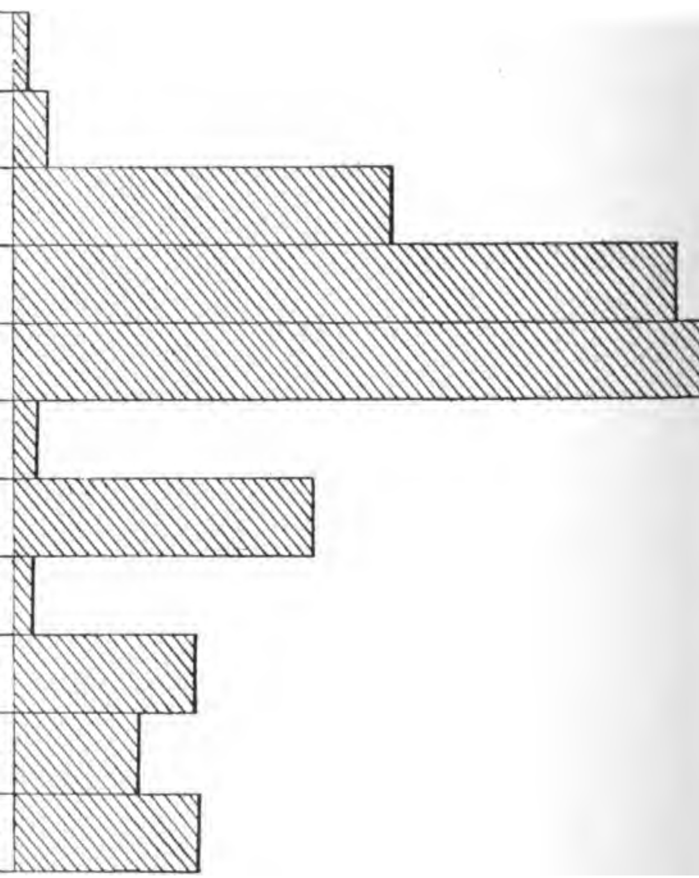
* This one had pits due to defective nickel plating.

TABLE II.

Experiments with Steel Tubes by Messrs. Yarrow & Co.

Kind of Tube.	How Tested.		Loss per sq. inch per ten days in milligrammes.
All pieces from the same mild steel tube.	Sealed with boiled sea water and hydrogen ...		·11
	Sealed with boiled sea water and air		·30
	Sea water open to air ...		4·40
	Sea water, air bubbling through		7·85
	Sea water at 100° F. open to air		8·20
	Sealed with boiled town water and hydrogen ...		·19
	Town water open to air ...		3·46
	Sealed with boiled distilled water and hydrogen ...		·14
	Sea water at 10 lbs. pressure	This pressure was not constant.	2·13
	Town water at 10 lbs. pressure		1·47
	Distilled water at 10 lbs. pressure		2·14

Relative amount of loss per ten days shown shaded.



The general conditions of these tests were the same as for Condenser Tubes (Table I.).

Note.—The Scales in Tables I. and II. are not identical.

Mr. F. EDWARDS (Member): My Lord and Gentlemen, it does not appear to be stated in this paper whether the corrosion originates from the inside or from the outside of the tube. On page 220, under (c), I notice that the inside of the tubes is especially referred to. I have had some experience of this matter which, I think, may perhaps be of some use to the members. Many years ago I received a report from one of our engineers, stating that the tubes in a certain condenser were giving out very rapidly, and indenting for an entirely new set. I therefore asked him to send us some pieces of the defective tubes, and on receiving them, I sent them to Mr. Wilson, the analytical chemist. I have got some of them here. They are not from this particular ship, but we have had a good many ships that have given more or less trouble. The members will see from these pieces of tube that the corrosion has commenced from the inside. Mr. Wilson found oxide of iron lying in the bottom of these cavities: now, in the first place, the brass corrodes the iron, and, after that, the action is reversed, and the oxide of iron corrodes the brass. Immediately I got that, I looked at the drawings of the condenser, and I grasped exactly what was going on. The pit holes in these tubes are all on the lower side of the tubes—none of them are on the upper part. The condenser was horizontal, the water passing through it twice. It happened that, in this steamer, the circulating pump drew the water from the bottom, and the water was sucked through. The cold water came in at the top, and, as very often occurs in connection with condensers, the main circulating valve of the circulating pump was closed to a considerable extent. The pump emptied the condenser to a great extent, so much so that the water fell from the top into the lower portion of the top box, and went through the bottom tubes only, and then, at the other end, tumbled down into the bottom of the bottom box, and then went through the second set of bottom tubes. At the end of the first voyage, the builders thought they could remedy the evil by some arrangement for filling the top tubes. They put in a weir at each end, so that none of the water could escape until it got up to the level of the top tubes, and then it tumbled over the weir. They then did the same thing with the bottom box, so that all the top tubes in each box had to be full before the water could get to the circulating pump. The letters of the chief engineer gave a full description of what occurred. All the tubes that pitted were bottom tubes; those in the top box, and others in the bottom box, were all pitted near the end where the water entered. I came to the conclusion that the cold water coming in at the top caught and corroded the cast-iron jacket, and that particles of oxide of iron came off and then sank down. Some of them, no doubt, got into the top tubes, and they were sent on at a high speed to the other end. Others sank down, and were carried a certain distance into the bottom tubes, but the current through them was so very sluggish that it was unable to carry the oxide far into the tubes, and it was deposited on the bottom of these tubes, near the entrance. Then some more oxide may have been knocked off at the other end, and the particles that may have come through the top tubes in the upper box, in sinking at the other end, evidently got carried in, and came to rest in some of the lower tubes of the bottom box. I then felt practically certain that I knew my position, and that I could get over the difficulty. I simply wrote to the chief engineer and told him that the water in many of the tubes, and particularly the ones that had corroded, was travelling too slowly —so slowly that it was leaving the oxide of iron in. I sent him a copy of Mr. Wilson's report. The weirs that the builders had put in were of gun metal. The engineer grasped the position at once, and he had the weirs melted up and put in three divisions. The water then went through four times instead of twice. The result was that one or two tubes, which were on the point of failing before he made the alteration, gave out, but with the rest, there was no trouble whatever. That is not the only condenser with which I have had that similar experience. I have recently taken over the charge of several steamers, and the very same thing has happened. The engineers in each case complained of the tubes; we took them out, cut them up and examined them, and we found that in each case the trouble originated from the inside. If the condensers were designed for the water to go through twice, or if it has been doing so, the only thing we can economically do is to put it through four times, and, when we have done that, we have had no more trouble since. These samples are from one of Messrs. Wilson & Co.'s boats, of Gothenburg. If the trouble arises from the inside of the tube, you can rely upon it (at least so far as my experience goes) that increasing the speed of the water will put a stop to it.

Mr. A. Spyer (Member): My Lord and Gentlemen, it is a great source of gratification to all of us to find that foreign correspondents take the trouble to prepare and read before this Institution papers of such decided interest to engineers as this one is. The question of the corrosion of condenser tubes is somewhat mysterious, and the observed phenomena may be roughly divided into three classes. There are, firstly, defects due to manufacture; secondly, defects traceable to mechanical action or arrangements, and, thirdly, and most important, the chemical action, or pitting, which Professor Cohen has been investigating. The cause of this latter action is certainly not clear, and, up to the present, we have failed to find any entirely satisfactory explanation for it. In the Navy, I need hardly say, we have experienced these troubles with condenser tubes, and we might, from the nature of the case, expect rather more of them than the Mercantile Marine, because it is our practice, owing to considerations of weight, to make the condenser shells generally of gun metal. Consequently, there is a very much greater tendency to galvanic action between the shell and the tubes, than where the shell is of cast iron. On the other hand, it is Navy practice to put one per cent. of tin into the composition of the tubes, which is in favour of our tubes, as compared with those of the composition which the author gives, that is, 66 per cent. of copper and 33 per cent. of zinc. The Navy tube contains about 70 per cent. of copper and 1 per cent. of tin, which is in favour of the tube. Nevertheless, we are troubled with occasional instances of this pitting, which we are unable to explain satisfactorily. Of course, I am excluding the two classes of trouble before alluded to, those due to manufacture and to mechanical action. I may say, with reference to another remark in the paper, that we have tried aluminium bronze compositions. The effect of using this varies very much, according to what is meant by aluminium bronze: whether it is bronze containing zinc, or a pure bronze with copper and aluminium only. Naval experience with the latter material certainly does not coincide with the author's statement that aluminium bronze corrodes more rapidly; on the contrary, it has been found in some cases, where we have fitted condensers with tubes of copper aluminium, that corrosion ceased where we have not been able to explain its previous occurrence. In other cases it continued, but more slowly. If Professor Cohen can successfully trace the causes, our troubles would soon vanish; but these causes are by no means clear at present. There are many suppositions on the point, but, personally, I am certainly very much inclined to think that he has put his finger on the right spot when he says that electrical leakage is one of the most important causes. In modern ships, the use of electrical motive power is increasing every day, and this means more possibilities of electrical leakage. There is no doubt that the corrosion, not only of condenser tubes, but of copper pipes and of bolts and fastenings, has increased of late years, and is, at times, more rapid than was the case before. Whether we shall succeed in curing this trouble is quite another matter. As to the remedies which are proposed by the author, the tin coating is by no means new. Years ago, in the days of cast-iron condenser shells, we used to coat the tubes with tin. Then it was discovered that it was not much use coating the tubes with tin so long as the shells were of cast iron. We then came to gun metal, or metal shells, and the question was again considered: should the tubes be coated with tin? As a matter of fact, the difficulty here is very much a mechanical one. You cannot get a satisfactory coating of tin all over the tubes, and if you have a tube coated with tin with one or two bare spots, you are worse off than if you have no coating of tin at all. As to the oxide of copper coating, that is a thing we have not tried; but in the case of copper pipes exposed to scouring action, there is a very strong tendency to get spots of corrosion, or of oxide, in the copper, and the immediate result of these, in an otherwise clean pipe, is that the pipe perforates very rapidly at these spots. Whether you could get a complete coating of oxide of copper on a pipe or not, I do not know; but, as the author himself says, the practicability of the application is very much open to question. As to the nickel coating, there is very little experience to go on. I do not quite know how it is proposed to obtain the coating of nickel, whether it is to be deposited electrically, or merely by a mechanical arrangement. Experiments are being made in America with what are called nickel tubes—that is, not pure nickel, but tubes of about 80 per cent. of copper and 20 per cent. of nickel. I believe, however, that the experience there has not yet been sufficient to pronounce definitely as to whether anything is gained thereby. As to (*d*), the author suggests "an electrical connection between the copper or brass tubes and zinc or iron plates, which are suspended in the sea." There you have, I think, the only means that at present appear to be of any practical value, that is, the employment, in the condensers, of cast-iron or zinc plates in good metallic connection with the tube plates, and so placed that the circulating water passes as far as possible over those plates before it obtains access to the interior of the condenser tubes. Up to the present, this treatment has proved to be reasonably satisfactory; but, nevertheless, we are still troubled occasionally with mysterious cases of pitting. As to proposals (*e*) and (*f*), I do not quite understand the author. These conditions seem to me to be a little contradictory. Perhaps I may not be quite clear as to what is meant; but in one case it is suggested that the current should pass through the tubes, and in the other case that the tubes should be carefully insulated, and I do not know which is intended. In Navy condensers, which are secured with screwed ferrules and packing, the tubes are partially insulated in the tube plates. There are many points in this paper which, when investigated, may result in considerable advantage to our incomplete knowledge of the subject.

Mr. F. Tomlinson (Visitor): My Lord and Gentlemen, from time to time I have had to investigate a great number of cases with regard to the corrosion and pitting of condenser tubes. Some little time ago (on the instructions of my principals, the Broughton Copper Company) I drew up a general report on these various investigations, several extracts from which, with your permission, I will now read:—"In all cases of pitting and perforation that we have examined, it was found that the metal had been converted into oxy- or basic-chlorides of the metal or metals composing the tube, at the points where the corrosion had taken place. This is conclusive evidence that the active corrosive agent is chlorine. All condenser tubes in sea-going boats will corrode, and in those boats frequenting docks and harbours where the water contains much impurity both in solution and suspension, the corrosion and pitting are always active; in some cases especially so. In ships which avoid foul water, the corrosion of the condenser tubes is found to be comparatively slight, and it takes place fairly evenly over the whole surface of the tubes, the destructive pitting being largely avoided. It is also requisite, for tubes to give such favourable results, that they should be worked under favourable conditions of management. The condenser must not be subjected to any stray currents of electricity, and no particles of metal, metallic oxides, or carbonaceous matter must enter with the cooling water and be deposited on the surface of the tube. The galvanic action caused by such particles of foreign matter resting on the tubes decomposes the saline water by which the foreign particles are surrounded, thus liberating the active element chlorine, which speedily attacks the tube with the too familiar disastrous results. Such particles of foreign matter naturally accumulate on the lower interior surface of the tube, generally about the middle of its length, which is often covered with saline water even when the engines are at rest. This, in many instances, accounts for the corrosion being most severe on the lower interior surfaces of the tubes." The bad influence of oxide of iron, &c., inside condenser tubes, which sets up a galvanic action, has already been explained by a previous speaker (Mr. Edwards). "This galvanic and chemical action is greatly accelerated by heat: thus, we generally find the corrosion most severe in that part of a tube which has been unduly heated, for instance, by steam impinging directly on the tube. Where such corrosion has been traced to this cause, some alteration to the design of the condenser has prevented any further trouble in this respect. Further, if the condenser becomes connected to, or influenced by any electrical current, as is sometimes the case, the corrosion becomes more rapid." The effect of electrical currents on board ship has already been mentioned by Mr. Spyer in his remarks, and, in reference to this, I may quote a case in point of rapid failure of tubes which I investigated a few weeks ago. In this case, it was found, although we were assured that the ship was wired on the two wire system throughout—that is, a complete lead and return—yet on making a connection between the condenser and one of the mains of the dynamo, we got practically a dead short circuit, thus showing that there was what is known as a bad "earth" somewhere in the circuit. Anybody familiar with electrical wiring, knows that it is very easy to get an "earth" on the frame of a ship. "This acceleration of corrosion by electricity can be practically demonstrated experimentally, if a small piece of tube be partly or wholly immersed in salt water, and a current of electricity passed through the tube, or through the metal vessel containing the tube, when rapid corrosion will be set up." This is a simple experiment, and can easily be carried out, but it requires several weeks sometimes before this pitting takes place. A series of such experiments shows this to be the case. When tubes corrode rapidly and fail, shipowners are sometimes naturally inclined to blame the tubes; but we find, on inquiry, "that in such cases of rapid failure of the whole or part of a set of tubes, other similar tubes, amounting often to many thousands, give excellent results, yet all these tubes were manufactured concurrently from the same lot of castings and the same lot of finest materials, and all underwent exactly the same process of manufacture. The natural question then arises, why do not these other thousands of tubes fail in a similar manner?" I might mention that it is now possible to make a better tube by a new process, and that I have superintended the making of tubes by this process, in which the original fluid metal is subjected to great hydraulic pressure, and a hollow shell is forged from the solid billet by hydraulic pressure, by which means a denser and more homogeneous tube is obtained. Professor Cohen's experiment, showing the effect of zinc plates in the condenser, is very interesting. Zinc plates have been applied in practice for years; but he shows that there is a limit to the protective qualities of these zinc plates, as it is possible for the electrical resistence to get so great that the plates will not have the desired effect. Probably, this accounts for cases which have been experienced when these protective plates have been put in condensers, yet failed to eliminate the corrosion of the tubes. Professor Cohen recommends a thick coating of tin one millimetre thick on the tubes. An even coating so thick as this cannot be made by the ordinary dipping process, which leaves the coating only about $\frac{1}{50}$ millimetre thick; but it can be accomplished by mechanical means, and here I might remark that, as tubemakers, we are always most desirous to assist in the advancement of any suggested improvements. Acting on this desire, on the receipt of a copy of Professor Cohen's paper, I made a brass condenser tube with an inside coating of tin of one millimetre thick, a sample of which I have here present. With tubes tinned by the ordinary dipping process, as Mr. Spyer remarked, there would very often be bare patches inside a tube. The presence of such bare patches makes the tinning worse than useless. Then, with regard to the question of nickel and copper alloy, an alloy of 15 or 20 per cent. of nickel copper for condenser tubes has been mentioned. Tubes of this and other alloys can be better made by the process I have previously named—hydraulic forging. I have here one or two small samples of such tubes. Our thanks are due to Professor Cohen for his very interesting paper. It has not eliminated the mystery of corrosion and pitting altogether, but it leaves a field for further research in this direction. What appear to be required are conditions of working under which there will be less tendency for tubes of any metal to corrode or pit.

Professor Ernst Cohen (Visitor): My Lord and Gentlemen, I think it is not possible to answer all the questions that have been asked, but some of them I will endeavour to reply to. The first is that nickel tubes would be good for the purpose. I think a combination of nickel and copper would not keep, but I have heard that here in England nickel tubes are now made of pure nickel—nickel of 99·9 per cent. They are very expensive, costing, I think, seven or eight times as much as brass or copper tubes; but I know, for instance, of two ships which have been built at Rotterdam, and one of them has these nickel condenser tubes. One of the speakers said that aluminium bronze would not corrode so very strongly. I think he said aluminium bronze with some zinc in it. The aluminium bronze that I used was copper and aluminium—5 and 10 per cent. aluminium. Then it has been said that tubes with tin coating are used, but I think that, unless the tin coating is extremely thick, they will soon be corroded, and I think the best method with those tubes is to put on a very thick layer of the protecting metal. I do not know how it can be made in practice, but I have been told that it would be possible to put on a layer of tin of one millimetre in thickness.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G.): Gentlemen, I think you will join with me in a hearty vote of thanks to Professor Cohen for coming over here and reading this very interesting paper.

The following communication has been received from Mr. C. H. Wingfield (Member): The author has dealt very ably with one cause of failure of condenser tubes: viz., corrosion on the surfaces next the circulating water. Their external surface—that next the steam—used to be attacked a good deal by fatty acids from the exhaust steam, and, though the use of mineral cylinder oils has largely reduced this trouble, it is not altogether unknown, especially in those parts of the tubes under the exhaust branches and those near the tube plates. It is possible that the latter local action is due to steam and acids, condensed on the tube plates, trickling down, and so attacking the tubes near their ends. If so, a flange might be screwed into the plates round each tube, so as to prevent any wet on the plate trickling on to the tube. Another trouble is that tubes will sometimes split without any apparent reason, thus causing leakage of salt water into the feed system. As this occurs notwithstanding a previous hydraulic test, it is evidently due to some molecular change in the material. It occurs as often in tubes which have never been used as in those fitted in the condensers. I have been at some pains to ascertain, if possible, the conditions accompanying this change of structure, and I found it was always associated with damp in those instances which I was able to examine into. Brass tubes left standing against a wall will often be found after a few months to have split open at their lower ends if they are on damp earth, and, in one case, the split had extended throughout the length of a condenser tube so placed. Tubes of all sizes, from a $\frac{5}{8}$ in. condenser tube to a $\frac{1}{4}$ in. water-gauge drain, have been under my observation, and, in nearly every case, the split at the lower end developed in course of time. The tendency of brass wire to become "rotten" is well known, and this also appears to be in some way due to damp.

Brass sheets left standing on edge often have to have their lower sides sheared off before they can be put to any useful purpose. It would seem that condensers—like boilers—should be either wet or dry, but never damp, if tubes are used which are so drawn as to be in a condition of stress tending to open an incipient crack. Certain batches of tubes appear to have a marked tendency to crack, and it is probable that, where this stress is not present, the tube might indeed become brittle by the action—whatever it is—of damp, but that a crack would not form. Tubemakers could probably account for this condition of stress, which causes a tube, when cracked, to open out. It is probably due to the final draw producing more compression on the inner than on the outer surface. If so, a modification of the dies or mandrels used should cure this annoying tendency. There is one point in the paper about which a little further information seems desirable. The tubes in Fig. 4 (Plate XXIII.) are glass. Does the author mean to suggest (on page 219) that the action would have been the same, had they been of metal, as in the case of the circulating pipes of a ship? I fear the electric current would, in the latter case, take the shortest circuit, and confine itself to the circulating pipes, or to the nearest exposed surface forming a part of (or in metallic connection with) the metal hull, and that it would not take the trouble to go on its errand of protection to the condenser tubes at all, unless these are insulated from the condenser. Has the author obtained evidence from actual cases of the action described on page 219? In view of the above, it would seem difficult to so apply the suggestions (*d*) and (*e*) on page 220, as to give adequate protection to the tubes.

METHODS OF HANDLING MATERIAL OVER SHIPBUILDING BERTHS IN AMERICAN SHIPYARDS.

By William A. Fairburn, Esq., Member.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 21, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

Shipbuilding as an industry in the United States has made wonderful progress during the past few years. Ten years ago there were only two shipbuilding plants in America capable of building an armourclad or a very large merchantman, whereas now there are seven yards building big armourclads, and three other yards occupied with the construction of merchant vessels over 550 ft. in length. At the present time there are in successful operation twelve shipbuilding plants on the sea coast of the United States, each of which is capable of building the largest naval or merchant construction, and there are seventeen other well-established yards on the east and west coasts for vessels of average size. On the Great Lakes there are nine yards, capable of building the largest class of Lake vessels, and five other yards that can successfully build average Lake ship construction.

Previous to the construction of the Atlantic mail steamships *St. Louis* and *St. Paul*, built at the yard of the William Cramp & Sons Ship and Engine Building Company in 1893, no large merchantmen had been built in the United States; and until three years ago no other big merchant vessels, excluding the above American liners, had been contracted for, the three yards—William Cramp & Sons, Union Ironworks, and the Newport News Shipbuilding and Dry Dock Company—having had practically the monopoly of large vessel construction, which consisted principally in the building of nine battleships of from 10,250 to 11,500 tons displacement each. To-day, however, the biggest cargo carriers, passenger steamships, and naval vessels in the world are being built in America, and among the merchant vessels, seven large Atlantic and four Pacific liners are worthy of special mention. The construction has been spread and well distributed throughout the yards of the country. The older, well-established plants, capable of handling such vessels, have been taxed to their utmost capacity. Other shipbuilding companies have much improved and enlarged their plants, embracing the opportunities presented, and other new shipbuilding yards have been established to take their share of the ship construction work, the demand for which is the outcome of the expansion policy of the present Republican Administration. To-day, therefore, the conditions in the United States are vastly different from what they were a few years ago. There are a great many more shipyards, and the majority of these are now building vessels each weighing about three or four times as much as those built in America a decade ago.

The great increase in the number of shipbuilding plants in the United States during the past few years has caused keen competition, this being particularly evident on the Great Lakes previous to the consolidation of the largest Lake yards, and on the coast whenever Government work is advertised.

Until a few years ago the old method of erecting wooden derricks alongside the shipbuilding ways for lifting on board and erecting hull plates and shapes gave evident satisfaction, but of late years competition has necessitated more economical and more rapid methods of construction, and the increase in the size of vessels built, together with the corresponding increase in the weight of individual members of hull construction, has, in many instances, made a more approved system of handling material over shipbuilding ways an absolute necessity.

It may be well to here state that one of the causes for crane service over large shipbuilding ways in Europe, viz., the handling of power riveting tools, has not affected in the slightest degree the desire for overhead crane service over building berths of American yards. Whereas the largest hydraulic riveters that have been extensively used in European ship construction weigh about seven or eight tons, the weight of the present pneumatic tools used in American yards to do the same work is insignificant, and the most modern practice in the United States is such that power hoisting, lowering, and general handling of tools is practically never required. Deck and tank riveters, reamers, and drills are wheeled to their work. Very few yoke riveters are used, and riveting hammers, holders-on, chipping and caulking hammers, drills, reamers, &c., can all be readily carried by the workmen. No yoke riveters are being used on the largest and heaviest vessels now building in American yards. For instance, the Great Northern Steamship Company's two big vessels, building at the yard of the Eastern Shipbuilding Company, at New London, Connecticut, are being riveted throughout with jam and long stroke riveters. The vertical keel, longitudinals, and part of the frames have been riveted with jam riveting machines weighing only 36 lbs. each. Other parts of the frames, the bulkheads and casings, and the shell sides, have been riveted with long stroke hammers weighing only 24 lbs. each. The decks and tank tops have been riveted with the carriage riveters that weigh, all told, from 130 to 200 lbs., the weight of the carriage, which varies from 80 to 150 lbs., being included in this figure. With this machine there is no weight whatever on the operator, the machine being wheeled to its work. The bottom of the shell has been riveted with a jam shell riveter that weighs 147 lbs., all told, the hammer and ball weighing 66 lbs. With this arrangement also there is practically no load on the operator. The reamers and drills weigh about 38 lbs. each, and those fitted with angle attachment 44 lbs. each; the chipping and caulking hammers vary in weight from 9 to 11 lbs. each. The general method of ship riveting at the yard of the Eastern Shipbuilding Company is very similar to that being adopted in other American yards, for compressed air is the power used throughout the country for this purpose, and the weights above quoted are sufficient to prove that the handling of power riveting machines, &c., has had practically no influence on the need of crane service over the shipbuilding ways of American shipyards.

There are now in satisfactory operation in American shipyards the following types of crane or trolley service feeding the building berths:—

(1) Cantilever cranes.
(2) Gantry cranes.
(3) Gantry cranes with single cantilever extension.
(4) Gantry cranes with double cantilever extension.
(5) Ship house, roofed over, with crane service.
(6) Ship house frame, no roof, with crane service.
(7) Wire rope cable trolleys.
(8) Locomotive cranes, standard type, on tracks alongside berths.
(9) Locomotive cranes, derrick type, on tracks alongside berths.

The following is a list of the principal American shipbuilding plants, giving the type of crane service over shipbuilding berths in operation at each yard:—

American Shipbuilding Co., Lorain, O.	Brown electric gantry crane.
American Steel Barge Co., West Superior, Wis. ...	Wellman-Seaver cantilever crane.
Bath Iron Works, Bath, Maine	Ship house, roofed over, with crane service, and wire rope cable trolley.
Cramp, William, & Sons, Philadelphia, Pa. ...	Brown cantilever electric cranes.
Cleveland Shipbuilding Co., Cleveland, O. ...	Brown hand-power gantry cranes.
Crescent Shipyard, Elizabeth, N.J....	Swing boom derricks with steam winches.
Chicago Shipbuilding Co., Chicago, Ill.	Brown cantilever crane.
Craig Shipbuilding Co., Toledo, O....	Electric gantry cranes.
Delaware River Iron S.B. Co., Chester, Pa. ...	Swing boom derricks with steam winches.
Detroit Shipbuilding Co., Detroit, Mich.	Brown cantilever crane originally fitted at West Bay City yard.
Eastern Shipbuilding Co., New London, Conn. ...	Wire rope cable trolleys.
Fore River S. & E. Building Co., Quincy, Mass.	Ship house, roofed over, with crane service.
Globe Iron Works, Cleveland, O.	Swing boom derricks with steam winches.
Harlan & Hollingsworth Co., Wilmington, Del. ...	Gantry crane with cantilever extension.
Maryland Steel Company, Sparrows Point, Md. ...	Locomotive cranes, derrick type.
Moran Bros. Co., Seattle, Washington	Ship house with crane service.

Neafie & Levy S. & E. Building Co., Philadelphia.	Swing boom derricks with steam winches.
Newport News S. & D.D. Co., Newport News, Va.	Brown cantilever electric cranes.
New York Shipbuilding Co., Camden, N.J. ...	Ship house, roofed over, with crane service.
Risdon Iron Works, San Francisco, Cal.	No crane service decided on as yet.
Sewell, Arthur, & Co., Bath, Maine	Locomotive cranes, standard type on tracks alongside berths, also swing boom derricks with winches.
Townsend & Downey S. & R. Co., New York ...	Cable system proposed. At present swing boom derricks with steam winches.
Trigg, William R., Co., Richmond, Va.	Gantry cranes with cantilever extension building. Temporary wire rope cables now in operation.
Union Iron Works, San Francisco, Cal.	Ship house frame, no roof, with crane service.
Union Dry Dock Co., Buffalo, N.Y.	Wellman-Seaver gantry crane, with double cantilever extension.
Wheeler, F. W., West Bay City, Mich.	At time of consolidation of Lake yards, had one Brown gantry crane, with single cantilever extension, and one Brown cantilever crane. The latter has since been slightly modified and moved to Detroit.

NOTE.—Several of the above-mentioned Lake yards are now known, since the consolidation, as the American Shipbuilding Company; but in the above list it has been considered desirable to give the shipbuilding plants their old and better known names.

We will now describe the various methods of handling material in use in America over shipbuilding berths under the headings and divisions before mentioned.

(1) Cantilever Cranes.

At the plant of the William Cramp & Sons Ship and Engine Building Company, Philadelphia, is installed an equipment of Brown cantilever electric cranes, designed and built by the Brown Hoisting and Conveying Machine Company, Cleveland, Ohio. At present three complete cranes, covering six shipbuilding berths, are in operation, and a fourth to cover two other building slips is contemplated. Each crane is mounted on a trestle, built entirely of steel, efficiently trussed and braced. The columns are pinned at top and bottom to the bridge trusses and foundations. At the centre of each trestle are two main diagonal struts for bracing and holding the runway longitudinally. These are pinned together, and to the trestle at their apex, and to the foundations at their bases, thus making the centre of the trestle a fixed point from which expansion and contraction may occur in either direction; the pinned connections with the vertical columns, together with the omission of longitudinal diagonal braces between columns, allow flexibility and motion due to variation of temperature. The illustrations (Plates XXIV. and XXV., and Fig. 1, Plate XXVI., and Fig. 1, Plate XXVII.) show the general arrangement and construction of these cantilever cranes. In the panoramic view of the plant (Plate XXV.), Cranes Nos. 1 and 2 only are shown.

Under Crane No. 1, shown near the centre of the picture, are building the U.S. battleship *Maine* and the Russian battleship *Retvizan*. Crane No. 2, shown on the left, was erected between the large south merchant shipbuilding berths, and two 12,000 ton International liners, the *Kroonland* and *Finland*, are now being constructed under the same. Crane No. 3 has recently been built between two building berths at the north end of the yard, and Crane No. 4 has not yet been installed, but it is proposed to locate it between Cranes Nos. 1 and 3.

The following Table shows the relative sizes and capacities of these cranes:—

Crane No.	1.	2.	3.	4.
Length of trestle	547 ft.	725 ft.	664 ft.	582 ft.
Height under crane	93 ft. 6 in.	93 ft. 6 in.	95 ft.	85 ft.
Overall length of crane	202 ft. 10 in.	202 ft. 10 in.	168 ft. 10 in.	143 ft. 10 in.
Maximum beam of ships on ways	76 ft.	76 ft.	66 ft.	54 ft.
Gauge of track on trestle	20 ft.	20 ft.	14 ft.	14 ft.
Distance from crane down to track	33 ft.	33 ft.	22 ft. 9¼ in.	22 ft. 9¼ in.
Distance between columns fore and aft ...	41 ft.	41 ft.	41 ft.	41 ft.
Length of base of compression triangle ...	82 ft.	82 ft.	82 ft.	82 ft.
Lift at extreme travel of trolley	9,000 lbs.	9,000 lbs.	6,000 lbs.	8,000 lbs.
Maximum arm of crane	95 ft.	95 ft.	78 ft.	65 ft. 6 in.
Stated load...	25,000 lbs.	25,000 lbs.	20,000 lbs.	20,000 lbs.
Extreme designed arm to carry above load ...	60 ft.	60 ft.	46 ft.	39 ft. 6 in.

The carriage for each crane is driven by electricity; the motor, with necessary drums, gearing, and friction clutches for giving all the functions of the crane and trolley, is under the control of a single operator. The speeds of the cranes are all the same, being as follows:—

Hoisting full load	125 ft. per minute.
Hoisting one-third load ...	350 ft. per minute.
Hoisting 1,000 lbs. load ...	730 ft. per minute.
Trolley, transverse travel ...	400 ft. to 800 ft. per minute according to load.
Crane, longitudinal travel...	400 ft. to 700 ft. per minute, depending upon load and wind pressure. The minimum of 400 ft. per minute is with full load and against a pressure caused by a steady wind with velocity of 30 miles per hour.

For longitudinal track travel of the crane, power is transmitted from the electric motor, which is of 85 H.P. for the larger cranes, through bevel gears and shafting to the driving gears on the truck, which is controlled by suitable brakes and clutches. The transverse travel of the trolley along the cantilever arms is accomplished by the revolving of two drums winding wire rope connected to the trolley. These two drums are on one shaft, which runs always in the same direction. When one drum is thrown into gear with the driving shaft, the other is disconnected and remains idle, so that when one winds, the other "pays out." The drums are kept in the same relative position to one another by the use of a return loop or a small wire rope, which is wound in an opposite direction to the ropes connecting the trolley. The hoisting is accomplished by a single drum; the lowering is by gravity alone, governed by brakes. The resultant force of the crane under the load is kept within the proper limits by the use of a counterweight running on a track along the cantilever arms and above the hoisting trolley. It is connected by ropes to the trolley, so that it automatically takes a position on one arm of the crane, corresponding to the position assumed by the hoisting trolley on the opposite arm. For unusually heavy loads, an additional weight can be readily added to the normal counterweight. This dead balance weight can also have its arm adjusted when it is desired to hoist very heavy loads. This range adjustment is easily accomplished. When the trolley is brought to the centre of the crane, the counterweight automatically comes to the same transverse position. The weight is then disconnected from the ropes which connect it to the hoisting trolley, and the trolley moved out to a certain dimension, say 45 ft., on the crane arm, opposite to that on which the heavy load is to be lifted. When the hoisting trolley is in this desired position, the counterweight is re-connected to the trolley by means of the wire ropes above-mentioned. This operation occupies but a few minutes. If the heavy load is to be lifted at a point, say, 45 ft. from the centre of the crane, the readjustment of the counterweight will be such that, when the lifting trolley arrives at its point of hoist, the counterweight will in this case be 90 ft. from the centre of the crane on the opposite arm to the hoisting trolley; for, the travel of the trolley and counterweight being the same, the trolley has travelled from, say, 45 ft. out on the port arm to 45 ft. out on the starboard arm, a distance of 90 ft., while the counterweight has travelled 90 ft. out on the port arm of the crane.

The crane carriage is built with a large open platform for storing a load of plates, &c., while the crane is at the upper end of the ways near the plate shop or punch shed. When the load is aboard, the crane is moved down the trestle, and the trolley delivers each piece at its proper place. Heavy loads are picked up by the trolley and carried directly to their allotted place.

In the design of cranes, built for duty at the Cramp yard, wind area has been reduced as much as is consistent with strength and durability. The trestle has been built with posts 41 ft. between centres, leaving a free and clear space for storage and railroad tracks, and the intention has also been to build trestles that would not interfere with the handling of material in the vicinity of the cranes. The Cramp Shipbuilding Company operated their Crane No. 1 for a year before they placed the order with the Brown Company for Crane No. 2. Their continued orders for these cranes is a positive proof that they consider them very desirable and profitable acquisitions to their shipbuilding plant. Crane No. 1 had an unusually busy initiation to the Cramp yard, as it was called upon to deliver, not only the greater part of the material to the battleships *Retvizan* and *Maine*, but also a great deal of material to the Russian cruiser *Variag*, building on the next set of ways, the material being pulled over to the cruiser by auxiliary engines, as the material was suspended from the crane hook. The stern post of the *Retvizan*, weighing 18 tons, was placed in position in twenty minutes by the Brown crane above mentioned, this period of time including taking it from the cars at the front end of the yard. After the launch of this battleship, the same crane, with but a few men, cleared the ways in four hours.

At the yard of the Newport News Shipbuilding and Dry Dock Company, Newport News, Virginia, there are six sets of building ways, between each pair of which is a trestle carrying an electric cantilever Brown crane, which supplies the required material to the two building berths between which the crane runway is located. These cranes were also designed and built by the Brown Hoisting and Conveying Machine Company, and are of the same general type as the Cramp cranes. The following dimensions and data concerning these three cranes may be of interest :—

Length of trestles	733 ft.
Height under cranes at after end	110 ft. 6 in.
Over-all length of cranes	187 ft. 4 in.
Gauge of track on trestles	20 ft.
Distance from cranes down to track	32 ft.
Lift at extreme travel of trolley	9,000 lbs.
Maximum arm of crane...	89 ft.
Stated load	25,000 lbs.
Extreme designed arm to carry above load	55 ft.

The speeds of the above-mentioned cranes, which are illustrated on Plate XXVI., Figs. 2 to 6, and Plate XXVII., Fig. 2, are as follows :—

Entire crane along runway, with load of 9,000 lbs., 89 ft. out from centre, 750 ft. per minute.
Entire crane along runway, with load of 25,000 lbs., 55 ft. out from centre, 690 ft. per minute.
Hoisting full load—25,000 lbs.—200 ft. per minute.
Hoisting empty hook, 400 ft. per minute.
Trolley transverse travel, 400 ft. to 800 ft. per minute.

The Newport News Company have a fourth Brown cantilever crane, which was originally operated upon a wooden trestle in the space now occupied by the new dry dock. The trestle was removed, building berths abandoned, and, as yet, the crane has not been re-installed for service over building ways. This pioneer crane was steam driven and possessed the following characteristics :—

Height under crane	75 ft.
Over-all length of crane...	136 ft. 4 in.
Gauge of track on trestle	14 ft.
Distance from crane down to track	16 ft.
Lift at extreme travel of trolley	6,000 lbs.
Maximum arm of crane	64 ft. 6 in.
Stated load	20,000 lbs.
Extreme designed arm to carry above load	30 ft.

The trestles forming the runway of the Brown cantilever cranes at Newport News are of interest. There are three at present in successful operation. One is built of wood throughout, trussed and braced by struts and diagonals, but the other two have been built of steel by the American Bridge Company, of Edge Moor, Wilmington, Delaware. These steel trestles are different from the Brown designed and built trestles, such as are fitted at the Cramp and other yards, inasmuch as the tracks are carried by solid plate girders 72 in. deep by $\frac{3}{8}$ in. thick, instead of the usual trusses below the tracks. The Newport News trestle girders are stiffened vertically by 4 in. by 3 in. by $\frac{5}{16}$ in. angle bars, with 5 in. by $3\frac{1}{2}$ in. by $\frac{7}{16}$ in. bars at butts. The top and bottom angles are 6 in. by 4 in. by $\frac{1}{2}$ in. double. These girders are placed vertically, and held to their work 20 ft. apart, at both top and bottom, by $3\frac{1}{2}$ in. and 4 in. angle bars $\frac{5}{16}$ in. thick, diagonally braced with 4 in. by 3 in. by $\frac{5}{16}$ in. bars. These same distance pieces and trusses are braced by 3 in. by 3 in. by $\frac{5}{16}$ in. angle bars. There is in the centre panel of the trestle, under each girder, a large A brace extending upward from the foot of one column to the centre of the panel, and down to the foot of the next column, somewhat similar to the Brown trestles, but in this case the compression struts are formed of 12 in. by $\frac{7}{16}$ in. steel plate, backed with four 6 in. by 4 in. by $\frac{7}{16}$ in. angle bars. At the forward end of the trestles a wide-spaced panel is located, the base measuring 69 ft. instead of 41 ft. 6 in. as elsewhere. This forward panel is also diagonally braced with compression members, formed of 24 in. by $\frac{1}{2}$ in. plates and four 6 in. by 4 in. by $\frac{3}{4}$ in. angle bars. The trestle columns consist of angle bars 3 in. by 3 in. by $\frac{5}{16}$ in., held 22 in. apart by means of flat diagonal bars. These trestles are well braced. The length of the trestles from centre to centre of end columns is 733 ft., and the height above ground on the forward end is 70 ft. 6 in., and on the after end 78 ft. 6 in. The length of the base of the compression triangle in the centre panel is 83 ft. The columns rest on, and are efficiently secured to solid piers of masonry capped with steel plates.

In this connection, it may be well to briefly mention the steam rotating and travelling Brown cantilever crane, which has proved so valuable in the material storage yard. This crane is illustrated on Plate XXVI., Fig. 5. It has a capacity of 6,000 lbs. on a maximum arm of 65 ft. The over-all length of crane is 131 ft.; height of crane above track, 30 ft.; width of track, 18 ft. 10 in.

The plant of the American Shipbuilding Company, South Chicago, Illinois (formerly Chicago Shipbuilding Company), is equipped with a Brown steam cantilever crane, of the general type above described and illustrated. In this yard vessels are built on a level keel, or flotation line, and launched sideways. Figs. 3 and 4, Plate XXVII., show the general arrangement and location of the crane in relation to one of the building berths and the punch shed. This crane has a capacity of 20,000 lbs. on a 25 ft. arm. On its maximum arm of 57 ft. 6 in., the designed full load is 6,000 lbs. The over-all length of crane is 119 ft. 10 in.; the height of trestle above ground is 32 ft. 4 in.; distance from under side of crane down to runway track is 16 ft., making the crane 48 ft. 4 in. above the ground; the gauge of track is only 11 ft.

Another cantilever overhead travelling crane of the Brown type was fitted a few years ago at the yard of F. W. Wheeler Company, West Bay City, Michigan, and was placed in the same line as, and end to, a Brown gantry crane with single cantilever arm—this latter crane will be described later. When the large Lake yards were consolidated, and the American Shipbuilding Company formed, the West Bay City cantilever crane was moved to the plant of the Detroit Dry Dock Company, Detroit, Michigan, and its overhang increased. This crane, as originally fitted at the Wheeler yard, was operated by steam and had the following characteristics:—

Over-all length of crane	144 ft. 8 in.
Maximum arm of crane	67 ft. 6 in.
Height of crane above runway track	15 ft. 10 in.
Height of crane above ground	59 ft.
Gauge of track	15 ft.
Designed load on maximum arm	8,000 lbs.
Stated load on arm of 42 ft. 6 in.	25,000 lbs.

Since the crane was moved to the Detroit yard the arm has been lengthened one panel, and the capacity of the crane at extreme arm has been reduced to 6,000 lbs. The crane is now operating on a wood trestle 500 ft. long, which was moved from West Bay City with the crane. This wood trestle, together with the original crane, is shown on Plate XXVIII., Fig. 1.

The crane service of the American Steel Barge Company, now known as the American Shipbuilding Company, at West Superior, Wisconsin, illustrated on Plates XXVIII., Fig. 2, and XXIX., Fig. 1, was designed by the Wellman-Seaver Engineering Company, Cleveland, Ohio, and the structural work was built by the Shipbuilding

Company. This system consists of an elevated steel track, or runway, extending the full length of the shipyard between two building berths, and mounted on this runway is a cantilever crane, which traverses the full length of the runway. By means of this arrangement, the material is transferred from the end of the yard, where it is delivered from the shops, to its place on the vessels with a very small amount of handling. The runway is of a very massive type, 420 ft. long, 36 ft. high, with 18 ft. crane track gauge. It is braced to withstand crane stresses. The lower portion of the runway is arched to provide for a railroad track down through the centre. From this track the material can be picked up by the overhead crane, and delivered where desired on the ships. A point of interest in the construction of the crane is the double runway between which the wheels of the crane travel, and by means of which the overturning of the crane on the runway is prevented. The crane is 170 ft. long from end to end of arms, 55 ft. high from top of rail to underside of crane, and has a lifting capacity of 30,000 lbs. half way, or 40 ft. out on the arms, and 10,000 lbs. at the extreme end of the arms, which is 80 ft. out. The crane has a lifting speed, full load, of 30 ft. per minute, and a traverse of the trolley on the crane of 400 ft. per minute, and of the crane upon the runway of 300 ft. per minute. It will be noticed that the vessels built at West Superior are launched sideways, the runway being midway between, and parallel to, the launching creeks. The runway is, therefore, designed to supply permanent scaffolding for one of the sides of each of the vessels building. This crane service has given excellent satisfaction. Electricity is the motive power adopted.

(2) Gantry Cranes.

Gantry cranes without cantilever extensions are not popular with American shipbuilders. If the trestle work of a crane is built of wood, it is natural that the trestle should form scaffolding and staging uprights, therefore an extension on the gantry, so as to hoist material clear of the trestle work and staging, becomes necessary. If the runways of the cranes were of steel or wood, placed so far from the vessel building that staging uprights had to be placed between the trestle and the ship, then a cantilever extension would not be necessary, or even desirable, providing that sufficient space was left for hoisting material between the trestle and the staging uprights, but such an arrangement would be expensive, and occupy much valuable room. The Harlan & Hollingsworth Company, many years ago, built a gantry crane and operated it upon wood trestles in that form for many years, but they have recently added to it a very desirable cantilever arm. The gantry cranes with single cantilever extension, now in use at the yards of the Craig Shipbuilding Company, American Shipbuilding Company, and William R. Trigg Company, are practically gantry cranes, the cantilever arms being very short. All these cranes, however, will be described under the next division.

(3) Gantry Cranes with Single Cantilever Extension.

At the yard of the Craig Shipbuilding Company, Toledo, Ohio, there are in successful operation two electrically operated gantry cranes with short single cantilever extension, the extension arm being merely sufficient to hoist steel material clear of the crane runway on the side opposite to the building berth. The length of the trestle, upon which these gantry cranes operate, is 450 ft., the span of the gantry is 56 ft., the height of the truss of the crane above the ground is 51 ft., and the distance from ground to the top of runway is 35 ft. The gantry track is horizontal and perfectly level. The vessels are constructed with the keel level, or with whatever drag they are going to have at their designed line of flotation, and all are launched sideways. Fig. 3, Plate XXVIII., Figs. 2 and 3, Plate XXIX., and Figs. 1 and 2, Plate XXX., show the general design of these cranes, and also one of the cranes installed over a steamer building for San Francisco owners, and the launch of the steamer *Buckmar* built for Boston owners. With this type of gantry and trestle work it will be noticed that the outboard trestle work and track must be removed at the time of a launch. The view of the launch of the *Buckmar* shows the trestle work and staging lying in the water. The Craig Company state that the removal and re-erection of the trestle is not an expensive matter, they having found that the entire cost does not exceed $75 for a 250 ft. vessel, and some staging uprights have always to be removed before a launch, even if no crane service is installed. The trestles are erected on piles and topped by 5 in. by 12 in. timber, with a 3 in. opening between them, so that the spauls and bearings for the staging can be placed at any height by adjustable bolts. The stringers topping these uprights are 12 in. by 14 in., fitted with 30 lb. railroad bars.

The crane has a track gauge of 8 ft. At the base the trestle spreads to 12 ft. The outboard trestle is 6 ft. wide throughout. The gantries were designed and built by the King Bridge Company, of Cleveland, Ohio. The designed capacity of these cranes is 10,000 lbs., but, as the majority of the loads to be handled range from 1,000 to 4,000 lbs., it was desirable to build cranes so that the lighter loads could be handied much more quickly than the heavier loads. With this in view, the fall block and carriage is built so that the rope reeving could be readily changed from two-part line to four-part line. The crane travels along the trestle at a speed of 200 ft. per minute. It has a conveying speed of 600 ft. per minute, and a hoisting speed on four-part line of 50 ft. per minute, and on two-part line of 100 ft. per minute. All the movements of each crane are controlled by one 30 H.P. motor, located in operator's house, the motion of crane along the trestle being accomplished by means of a clutch. The levers are all assembled in one rack, so that the operator can have a view of the fall block at all times. The approximate cost of the truss crane, motor, blocks, and wire is $10,000. Mr. John Craig, President of the Craig Shipbuilding Company, speaking about these cranes, said :—" We think that this is the best kind of crane to be used, particularly if the vessel it covers is to be launched sideways. After using this type of crane for nearly ten years, we do not know of anything that is equal to it for the purpose, and the probability is that in the near future we will build another crane for the same trestle, whose capacity would not be over 3,000 lbs., as in the construction of Lake vessels we have so many lifts of about that weight and less, that one heavy crane is hardly sufficient to do all the hoisting that is to be done on one of our 6,000 ton Lake vessels, and yet launch her in four months from the time the keel is laid."

There is a Brown electric gantry crane over each building berth at the plant of the American Shipbuilding Company, Lorain, Ohio, of the same general type as that in successful operation at the Craig yard, Toledo. There are three longitudinal tracks, comprising two runways, upon which the entire crane travels, two tracks on a trestle on one side of the ship's berth, and one track on a trestle on the other side. The mechanism for hoisting, travelling, and trolleying is in the house carried on the pier of the double track end of the crane. The clear span of these structures between supports is 58 ft. 6 in., and the cantilever extension is 14 ft., making the total length of the bridge tramway on which the trolley runs 72 ft. 6 in. The working capacity of these cranes is 14,000 lbs., and they will hoist this load at the rate of 30 ft. per minute, and will trolley across the bridge with full load at 250 ft. per minute, while the whole crane, with full load, can be travelled along the tracks at from 300 ft. to 400 ft. per minute. A single electric motor is used for each crane, operating through friction clutches, as in the ordinary machinery of this type. The operating levers are arranged in the upper part of the house, and so located that one operator can control the different motors at one time, if desired. These cranes are so arranged that the span can be reduced by units of 2 ft., by disconnecting the single pier and moving it, together with its supporting track, 2 ft. nearer to the double pier. This is desirable in places where the ships are launched sideways, as is usual on the Great Lakes, and where the outer trestle is movable and used for scaffolding, or as staging uprights, as well as a support for the crane. This crane is illustrated on Plate XXXI., Fig. 1.

At the works of the American Shipbuilding Company, Cleveland, Ohio, better known as the Cleveland Shipbuilding Company, hand-power gantry cranes with short cantilever arms, similar in general characteristics to those before described, have been giving satisfaction for years. These cranes were in satisfactory operation when the Cleveland Shipbuilding Company's new plant at Lorain was built, and the hand-power Cleveland gantries are really the forerunners and experimental predecessors of the Lorain electric gantries. The Cleveland gantry cranes, which are illustrated on Plate XXXI., Fig. 2, were designed for a capacity of 3,000 lbs., with a span between supports adjustable from 46 ft. to 53 ft. The tramway, over which the whole crane travels, can be of any required length. The mechanism for hoisting is fixed on a platform carried on two tracks under one end of the bridge, and upon the platform the operators stand. Steel wire ropes are used throughout. Provision is made for the front sheer leg, running on a single track, to tip or tilt to accommodate itself to any changes in alignment of the track, without increasing the friction or straining the crane. The whole crane may be run on a curved track. There are four speeds for hoisting, four for travelling across the bridge, and two for moving the load and crane along the tracks on which it is mounted. The entire crane on longitudinal tracks with load has a speed of from 70 ft. to 120 ft. per minute. The trolley speed across the gantry bridge is from 100 ft. to 200 ft. per minute. Two men can hoist 2,000 lbs. at a speed of 10 ft. per minute.

The William R. Trigg Company, of Richmond, Virginia, are now installing at their yard three gantry cranes with short single cantilever extension arms, exactly similar in type and general design to those described at the Craig Shipyard. These cranes are electrically operated. They will run on wood trestles, which form the side staging scaffolding, and, as all the Trigg vessels launch sideways, the outboard trestle work will be removed when each vessel is launched. The trolley on the cantilever crane arm will hoist the steel material or load to be lifted from flat cars, which will be moved by locomotive cranes on railroad tracks parallel to the building berths. Two of these gantry cranes will have a span of 50 ft. each, and a trestle track length of 500 ft., the third gantry having a span of 60 ft., and a trestle length of 700 ft. Each crane will have a designed capacity of 14,000 lbs., a hoisting speed, light load, of 500 ft. per minute, and a speed of crane travel along trestles, loaded, of 300 ft. per minute. It is interesting to note that the Trigg Company operate the only shipbuilding yard of any size on the sea coast of the United States where the vessels built are launched sideways. They have been driven to the Lake custom on account of location of their plant and the depth of water; therefore, it is but natural that the crane service in the Trigg yard should be very similar to that which has been found to be so satisfactory among Lake shipbuilders.

Several years ago the Harlan and Hollingsworth Company, of Wilmington, Delaware, fitted an iron gantry crane travelling on wood trestles over their large No. 3 ways. It was originally intended to make the trestle, or framework supporting the gantry track, so that the width or span of the crane could be increased should occasion arise; but as such a change necessitated other serious changes in the gantry of a more expensive nature, the idea was abandoned. The gantry crane has recently been improved by the very desirable addition of a cantilever arm, well trussed and braced, as shown on Plate XXXI., Fig. 3. The crane is electrically operated, and has a speed of

150 ft. per minute on the track, and 50 ft. per minute hoist. The designed dead load capacity of the gantry is 10,000 lbs., and of the jib or cantilever hoist 5,000 lbs. The runway for the crane is 500 ft. long and 50 ft. track gauge. The front posts, those directly under the track, are 5 in. by 8 in. timber double, with 3½ in. between them. The back posts are 4 in. by 8 in. The diagonal braces are 8 in. by 3 in., and the cross strut 8 in. by 8 in. yellow pine, all firmly bolted together. The sills on which the track is laid are 16 in. square. The whole structure is built on piles 14 ft. centres each way. The cantilever arm extends out 45 ft. beyond the gantry proper; the distance of the gantry track above ground is 45 ft., and the distance from the track to the hook of the crane is 25 ft., making the total height of the hoist 70 ft. from the ground. This gantry crane, as originally built, was used for many years with success, but in this modified form it is said to give perfect satisfaction. The Harlan and Hollingsworth Company launch their vessels stern first, or "end on."

One of the best examples of a real gantry crane, with single cantilever extension, is that now in operation at the American Shipbuilding Company's plant at West Bay City, Michigan. This crane was designed and built by the Brown Hoisting and Conveying Machine Company twelve years ago. It is steam driven, and runs on a wood trestle. The steadying end of the gantry travels on a track close up to the punch shed's open side, while the main gantry crane track is on a trestle close to the building berth, the trestle framing forming staging supports. The crane runs the length of the punch shed, picking material up from any part of the shop front, and trolleying it to any part of the ship, being built in the berth that is covered by the cantilever arm of the crane. Only one building berth is covered with this arrangement, and the vessels launch sideways into a creek. Plate XXXI., Fig. 4, shows the general arrangement of the crane, shop, and building berth.

(4) Gantry Cranes with Double Cantilever Extension.

At the shipyard of the American Shipbuilding Company, originally the Union Dry Dock Company, Buffalo, is a gantry crane with double cantilever extension, one end covering a building berth, the other end covering the plate shop, or punch shed. This crane service was designed, built, and installed by the Wellman-Seaver Engineering Company, of Cleveland, Ohio, together with the runway upon which it traverses. This crane is shown on Plates XXX., Fig. 3, and XXXII., Figs. 1 and 2. The Union Dry Dock Company's yard has only one building berth, from which ships are launched sideways into a creek or basin. Parallel with the building berth is the storage and working yard. The crane, it will be seen, extends over the building berth and the storage yard, and covers the plate shop, which is also parallel with the ship berth.

Material can be lifted out through the roof of the shop and transferred directly over to the ship upon the stocks. This construction leaves the berth free for sideway launching. This crane has a capacity of 30,000 lbs. on an arm of 40 ft., and 10,000 lbs. with an arm of 68 ft. The total length of the arms is 170 ft., the maximum length of traverse being about 160 ft. The maximum arm of the building berth cantilever is 68 ft., and that of the plate shop cantilever 43 ft.

The runway is composed of two separate well-braced tracks, with centres 49 ft. apart, the top of the plate girder runway base for the track bar being 30 ft. above the foundation. The height of the crane above the ground is 55 ft. The total length of the runway is 408 ft., the vertical supports for the same being 24 ft. apart. The crane body is of rectangular section throughout, the vertical sides being composed of steel plate, these sides being well tied and braced with parallels and diagonals. At the runway tracks, the crane is increased in width to the base in a V shape, the distance from centre to centre of the forward and aft set of wheels on each runway being 36 ft. The crane has a lifting speed with full load of 30 ft. per minute, and with a light load of 63 ft. per minute. The traverse of the trolley on the crane is 400 ft. per minute, and of the crane upon the runway 300 ft. per minute. This crane has been very satisfactory, and in constant service since it was installed. The arrangement, it will be noticed, is very similar to that of the West Bay City yard, previously described, but whereas the latter takes material from the ground at the front of the shop, the Buffalo crane has an additional cantilever arm, which takes material from the shop floor through hatches in the roof. By this method it is claimed that much of the general handling of material in the plate shop is eliminated. Fig. 3 (Plate XXX.) shows the stop placed on the end of the gantry crane track. This is a necessary feature, for one Lake gantry crane, not so fitted, was blown off the end of its track a few years ago.

(5) Ship Houses—Roofed Over—With Crane Service.

The Bath Iron Works, of Bath, Maine, built a steel ship house a few years ago in which to construct high-class vessels. This shed was roofed in, but had open sides, and the side posts were inclined outward, spreading towards the base. This shed is a light steel structure with wood roof. The posts are 12 ft. apart, and consist of angle-bars trussed with flat bars. Three longitudinal stringers of similar construction keep the posts in position. There are twenty-six bays, making the shed 312 ft. long, the width of the roof truss being 58 ft., with a width at base of 62 ft. Upon the roof of the shed is secured a timber framework, which supports a double Z bar track, in which move the carriages of the radial arc wire rope trolley system, which, as mentioned elsewhere, is in operation at this yard. This construction above the ship shed roof, together with plans of the ship shed, are shown on Plates XXXII., Fig. 3, and XXXV., Figs. 1 and 2. During the past year, an electric travelling crane of $2\frac{1}{2}$ tons capacity has been installed under the roof truss in this shed to handle material over the building berth. The crane runs on tracks having the same pitch as the building, viz., $\frac{3}{4}$ in. to the foot, and it is counter-weighted to allow for this gradient.

The Bath Iron Works also build small light craft, such as torpedo boats, under cover, the east end of the plate shop being specially arranged for this purpose. Four years ago, a 90 ft. single story extension was added to the 250 ft. two-story steel plate shop, and 150 ft. of the 340 ft. shop was reserved for light hull construction. This shop is 60 ft. wide, and built in a substantial manner. Plate XXXII., Fig. 4, shows the United States 30-knot torpedo boat, *T. A. M. Craven*, being prepared for launching from this shop. The machinery has always been put in position and the boats practically completed before launching, a portion of the roof being portable. The roof truss and mould loft floor girders—the latter occupying the west 60 ft. of the ship shed—are used in conjunction with ordinary blocks and falls for hoisting material on board these small and light vessels.

At the new shipbuilding plant of the Fore River Ship and Engine Company, Quincy, Massachusetts, there is now being constructed by the Boston Bridge Works, from designs of the Wellman-Seaver Engineering Company, a very interesting ship house 480 ft. long, 325 ft. wide, with height of rafters at the side varying from 100 ft. at the water end to 87 ft. forward. This shed, as shown on Plate XXX., Fig. 4, is of steel construction, roofed in, but with open sides and large skylights in the roof. In the centre of the house roof, running throughout its entire length, is a large trunk skylight, 46 ft. 3 in. wide, with ventilator slat or blind sides 9 ft. 6 in. high. Between the shed side posts and the tank skylight on the roof is a continuous skylight 25 ft. wide on each side. The central bay of the building is 185 ft. wide, which is ample width to take two of the largest vessels built or projected. The side posts are placed 60 ft. centres, and 10 ft. wide athwartship, tapering in a fore and aft dimension as they near the roof truss. This truss overhangs the posts 60 ft. in the clear on each side of the building, this overhang enabling two other good-sized vessels to be built under the roof, thus making in all four building berths under one roof. The sides are perfectly open. There are two 5 ton travelling cranes to be installed over each building berth, one being located for the port and one for the starboard side of each ship. There will be, therefore, eight 5 ton cranes of rapid action under this single roof. These cranes will be made by the Morgan Engineering Company, but as this construction has not progressed very far at the time of writing, further particulars are not available.

As before stated, there have been a large number of well-equipped shipbuilding plants built in the United States during the past few years. By far the most interesting of these, on account of the great amount of capital invested, and the elaborate shop and building slip construction, equipment, and crane service, is the New York Shipbuilding Company, located at Camden, New Jersey, on the Delaware River, opposite Philadelphia. This shipbuilding plant is practically under one roof, the machine shop, boiler shop, smith shop, plate shop, angle shop, mould loft, bending slabs, furnaces, and storage racks for steel material, &c., together with three large shipbuilding berths and a wet dock, being all connected together and practically made into one huge building. This building is of steel construction with back curtain walls.

A sketch is appended, Fig. 1, Plate XXXIII., which shows in a general way the scheme and lay-out of the shops and building berths. It will be noticed that the ship sheds extend from the wall of the machine shop towards and over the water. A basin or wet dock for fitting out the ships is in the first bay, thus allowing a large vessel to enter the basin, and be under a roof with its accompanying crane service during the fitting-out period. The three adjoining bays are each capable of taking either one or two sets of ground ways, one when a large ship is to be laid down, and two when smaller ships are to be built. Thus six vessels of moderate beam can be built at the same time. There are two 10 ton travelling cranes over each of these building bays, the inner end of the cranes being supported by a runway suspended from the roof. These cranes are placed high enough to clear any ship while building, or during the fitting-out stage when in the wet dock. The largest travelling crane in the works, which is of 100 tons capacity, Sellers' make, is here located, and so arranged that it can be used in any one of the four bays. This crane has a span equal to that of the building bays and the wet dock. It has ordinary crane travel throughout the length of the bays, and the lofty roof of the ship houses, embracing as it does the balcony of the machine shop, permits the hoisting of machinery from the shop below.

This large crane has a transverse carriage motion over the machine shop balcony and at the head of the building slips, so that it can be made to cover any portion of the machine shop balcony, and either of the ship house bays. Hatches are arranged over the machine shop balcony and in the gallery floor to facilitate the rapid handling of heavy weights, and engines and boilers can be picked up from the shop floor and placed where desired in the vessels, both before and after launching. Other cranes, known as ground cranes, feed material from the plate and angle shops to the building berths. These cranes, although of the overhead type, are below the keels of any vessels building, on account of the pitch of the blocking and ways. At present there are being constructed in these houses the following ships:—Two of 360 ft. length, one of 470 ft., and one of 500 ft. The two 360 ft. ships are building in the same bay, and the other two vessels occupy the only other two available bays. Besides these vessels, the company has contracted to build one other 500 ft. ship and two 600 ft. vessels. Adjoining the south end of the ship house and shops an extra set of ways with wooden trestles and travelling cranes has been built, which, it is stated, will only be used when all the other berths are occupied. Upon this set of ways, which are not roofed over, one of the 600 ft. vessels for the Atlantic Transport Company is at present being laid down.

Moran Bros., of Seattle, Washington, are making extensive improvements at their plant on the Pacific coast, and a large ship shed to cover the battleship lately contracted for is now under construction at their works. The shed, a section of which is shown on Plate XXXIII., Fig. 2, is 850 ft. long, 87 ft. 2 in. wide in the clear, and 66 ft. high under roof trusses. The sides and roof are completely covered, the side posts being placed 12 ft. apart abreast the ship ways, and 36 ft. apart abreast the side connections to shops. Two 5 ton cranes of equal length traverse the whole length of the house. In the design of this structure, the builders state that consideration was given to the manner of handling middle line material, and they concluded that such materials as enter into the middle line structure of vessels could be very advantageously handled by a side crane spanning half the breadth, the trolley of which travels within a short distance of the end of the bridge. It is their opinion that the advantage of having both cranes in unison, or either at will, independently available for middle line work, was greater than that of having one long crane to handle all middle line materials, reaching past the middle line, with a shorter crane at the opposite side.

In the Moran shed, provision has been made for the installation of an electric trolley over the middle line. This will be installed if they find that they have special use for a middle line crane, and require more hoisting units than the two cranes now about to be installed. The shed now being built is, as before stated, 850 ft. in length, but only one-half of the shed is taken up at the present time by the building ways, the remaining half being used as a ship fitting shop, with power tools arranged for the convenience of the work. This shed crosses the main shop building at right angles, which building is itself 650 ft. in length, and includes machine shop, boiler shop, and blacksmith shop. Electric travelling cranes of 30 tons' capacity pass without interruption across the ship shed at a considerable distance under the ship shed cranes. One side of the ship shed also forms one end of the fitting shop proper, where are located the scrive boards, furnaces, and bending slabs, this shop being 100 ft. wide, and the side of the ship shed is open for this distance. Two railroad tracks with main line connections also cross the ship shed within reach of the travelling cranes. The arrangement of the Moran Bros. shops is, therefore, somewhat similar to that adopted in the newly laid down New York Shipbuilding Company's Works, and their facilities for handling materials rapidly and economically are of the very highest order.

(6) Ship House Frame—No Roof—With Crane Service.

The Union Iron Works were one of the first in America to adopt, and put in successful operation, travelling cranes over shipbuilding berths. The system adopted by this firm consists of an extensive frame structure with posts and roof truss, spaced 12 ft. centres, not roofed in or enclosed. Overhead cranes travelling forward and aft are located under the lower chord of roof truss, and a series of independent jib derricks swing from the side framing, the uprights of which are utilised for staging purposes. The structure is not so elaborate as one is apt to consider from first impressions. It has most assuredly proved very efficient, and the vessels building always seem to be well lighted, free from draughts or resounding noise, which troubles are common with many roofed-in ship sheds. The Union Iron Works considered from the first that two cranes would be necessary for each vessel building, therefore all the ship sheds they have built are fitted with two cranes, with the exception of their latest and largest sheds, which are equipped with four travelling cranes. This company built in 1884 a skeleton ship house of timber, 300 ft. long, 48 ft. wide, with an average height of 55 ft., the top being built at an angle of ½ in. to the foot, sloping towards the water end. At the lower end was fitted a swing crane 35 ft. long; three years later a second ship house frame was built over berth No. 3. This house was the same length as its predecessor, but was made 85 ft. wide, 58 ft. high, and fitted with a swing crane 48 ft. long at each end.

Experience with these ship houses was so satisfactory that in 1891 the timber framework over berth No. 3 was duplicated over berth No. 4, the roof truss being made 3 ft. higher, and the end swing cranes fitted with 5 ft. more reach. Building slip No. 6 is now covered with a framework similar to those before mentioned, but much longer and higher. This skeleton house was originally 408 ft. long, 85 ft. wide, and 78 ft. average height; the sizes of materials are given in Fig. 3, Plate XXXIII. This frame structure has quite recently been increased in length to 480 ft., and the building of another skeleton house, the same length as No. 6, and 6 ft. narrower, is about completed over building berth No. 7. The general design of these ship houses, together with their location in the yard, is shown on Plate XXXIII., Figs. 3 to 5, and on Plate XXXIV. There is a swing crane at each end of these structures 50 ft. long, the larger ones each covering a vessel 580 ft. long, and, should it be desired to build still larger vessels in the shed, the construction is such that the trusswork can be easily lengthened, the height and width being ample to cover the largest vessels. The apex of the roof over ways Nos. 6 and 7 is 3 ft. from the centre, and one crane is made 6 ft. longer than the other. This is done to enable members of hull construction to be landed on the centre line of the vessel building. The overhead cranes are electric, and have a rate of travel fore and aft of 180 ft. per minute, an athwartship travel and a hoist of 90 ft. per minute, and a lifting capacity of 10,000 lbs.

The Union Iron Works prefer manilla rope for hoisting, on account of its elasticity, which enables a plate to be bolted up if placed by the crane within 1 in. or 2 in. of its true vertical position. For the erection of the plating under the bottom of the vessel it is interesting to note that a wire rope is used, rove through a hole in the plate, and the corresponding hole in the frame to be secured to, and well toggled under the plate, thus enabling the plate to be hauled up close to its place. The staging necessary for the construction of a vessel is of itself a large item in the cost. One feature of the Union Iron Works' structure is the rapidity with which staging can be erected. When the ship under construction is large, the spauls, which are 4 in. by 8 in., are rove through the main posts and held by loose bolts at the ends. When smaller or narrower vessels are built, a standard made of 3 in. by 6 in. double plank is set on top of the ground and held from canting or tilting by the rigidity of the spaul in the posts, thus saving all bracing. As the posts are of 12 ft. centres, plank for staging 26 ft. long, 12 in. wide, and 2 in. thick, is used throughout, and this has been found strong enough for any work, and light enough to be easily handled. The frame structure is also used to keep the upper works of the vessel fair during the early stages of construction, and turnbuckles are used from the sides of the vessel to the structure framework. This framework is well braced, and given longitudinal rigidity by means of diagonals, all of which is shown on the appended plates. The Union Iron Works find no disadvantage in having the posts as close as 12 ft., all the hoisting being done at the upper end of the berth, and the material carried over the vessel to the required place, where it is lowered into its proper position. As no side shores above the bilge are used, the top sides are always clear for lowering and taking plates into their true position.

The latest ship sheds have been fitted with 32 ft. riveting swing cranes, carrying 8 in. bulb tee trolley beams for pneumatic yoke riveting, but, as before stated, during the past year, jam and long stroke riveters have been gradually taking the place of the heavier pneumatic tools, and riveting cranes do not now seem to be necessary, or even desirable. The floor of these skeleton ship houses is constructed of timber, similar to a wharf, giving a fine level floor, which much facilitates erection and construction in general. It moreover affords a ready means of securing turnbuckles for holding a frame down and adjusting it, thus carrying out the San Francisco general method of fairing a ship on the sides down to the ship's bottom. The overhead framework here described has been used quite frequently for hoisting on board vessels undergoing construction weights which are heavier than the safe maximum load of the average cranes. Bed-plates, condensers, pumps, auxiliary machinery, and parts of main engines have been frequently put on board before launching, by means of falls connected to the rafters overhead. The present overhead cranes were built in 1891, operated by means of 4 in. circ. manilla rope, running about 3,000 ft. per minute. This rope ran continuously while the cranes were in use. There were three independent reversible friction clutches on a single shaft to control the hoisting bridge and cross travel, so that any and all of the functions of the cranes were available as required, but electric power has gradually been introduced through the building slip crane service. The driving sheave has been removed and replaced by a 40-horse power motor connected to the shaft, which still runs continuously, as with the old arrangement.

These ship sheds have been criticised on account of high cost when benefits obtained are considered, but it is no exaggeration to say that the wood framework and type of crane used can be built just as cheaply as other types of overhead crane service, whereas it has the great advantage of covering some additional desirable features, and having a larger number of hoisting units under operation. It is true that two cranes placed alongside of each other involve a noticeable increase in the strength of the roof truss, but the hoisting capacity is by this means doubled, with but a correspondingly slight increase in cost. Another criticism broached about this method of crane service is the fact that a small section in the width of the ship is not commanded by the overhead cranes. As before stated, the Union Iron Works cranes are arranged so that one of the two athwartship cranes centres the ship. The fact of a small longitudinal belt near the centre not being theoretically covered is not any serious objection, as only plates are practically placed in such a position, and these can be swung and left where desired. It has been said that the number and sectional area of the trusses for supporting the travelling cranes affect to a marked degree the lighting of the ship's berth. This is not so, for the ships are known to be well lighted during construction. Another objection raised is the fact that the side framing and foundations take up much valuable room, limiting the number of shipbuilding berths for a given width. The Union Iron Works have apparently all the room desired, and this criticism cannot be applied in their case. It is well to note, however, that other systems of crane service necessitate as much room between adjoining slips as the Union Iron Works system.

(7) Wire Rope Cable Trolleys.

Another method of handling material over shipbuilding ways is the overhead cableway or wire rope trolley system.

Wire rope cableways have been used in America quite extensively for years, and this means of handling material over great fields of service, with long spans, has long been considered by civil engineers the most satisfactory method, taking cost, utility, and time to erect, &c., into account.

In 1895 the Bath Iron Works, of Bath, Maine, built a small wire rope cableway for unloading material from cars, and depositing it in the storage racks of the steel material store yard. The fixed cable post and engine hoister were placed on the opposite side of the car track to the steel plate racks, and sufficiently remote to permit of material being hoisted from any part of an ordinary flat railroad car. The opposite end of the cableway was made movable by being secured to a car or set of rollers, which moved in a track laid as an arc of a circle for a distance of about 100 ft. The span of the cableway was about 200 ft., the diameter of the wire rope cable $\frac{7}{8}$ in., and the designed full load 1,500 lbs., with a factor of safety of 6. In a test of an hour's duration, 52 plates were unloaded from cars and deposited in steel racks with this apparatus. This system of handling steel material proved so very efficient, and the cost of installation and maintenance was so small, that the Bath Iron Works in 1898 decided to use the same system for handling material over the shipbuilding berths of their plant.

At that time three sets of ways were occupied intermittently by vessels undergoing construction. The ways were rather short and narrow, and only small naval and merchant vessels had been built up to that time. When the Company, in 1898, secured the contract for the construction of the pioneer American tramp steamship, now known as the *Winifred*, and the large steam yacht *Aphrodite*—both vessels over 300 ft. long—it was decided to arrange Ways No. 3 to take the heavier cargo vessel, and build a new set of Ways No. 4, with a ship shed over them, for the construction of the large yacht. The roof of the ship shed was built to take the radial arc track of an overhead cableway system, and a lofty steel spar, the fixed point of the cable system, with hoister house attached, was located alongside the plate shop near the main doors. The span was 282 ft.; the travel of the movable end of the cableway extended over a chord 320 ft. long; sag of wire rope allowed (with load), 10 ft.; diameter of cable, $1\frac{1}{2}$ in.; designed load, 1,500 lbs.; factor of safety, 6; carriage speed, 420 ft. per minute. The hoisting machine was operated by a 35 h.p. motor, 320 revolutions. During a test made, the system would hoist its load and run the length of the cable—some 280 ft.—in 52 seconds, and a series of tests, hoisting, travelling, depositing, and returning to the base of operations, averaged 1 min. 40 sec. The steel spar was 90 ft. long, and 36 in. maximum diameter.

Figs. 1 and 2 (Plate XXXV.) show the general arrangement of this cable system. Hatches with covers were placed in the roof of the ship shed, and the scheme worked very satisfactorily, taking all the material from the plate shop as machined, and locating it in position on the ships. For a small expenditure of money, the cable system did its work well, doing all that it was designed to do, and feeding Ways Nos. 3 and 4 in a satisfactory manner. For general shipyard use, however, where large and heavy vessels have to be built on each set of ways simultaneously, the scheme has the great disadvantage of being able to deliver material for only a small portion of the vessel's length on the ways near the fixed centre of the trolley system. For instance, whereas Slip No. 4 was covered for a length of about 320 ft., Ways No. 3 were covered for 265 ft., Ways No. 2 for 200 ft., and Ways No. 1 for only 135 ft.

The adoption of this wire rope radial arc or sweeping cableway system of handling material was assisted by the efforts of Mr. C. R. Hanscom, now president of the Eastern Shipbuilding Company, New London, Connecticut, who was connected with the Bath Ironworks as general superintendent at the time above-mentioned, when wire rope trolleys were constructed at these works. When the plant of the Eastern Shipbuilding Company was built in 1900-01, Mr. Hanscom adopted the plan of using wire rope cableways over the large building berths of this company, so as to give an effective rectangular field equal to that of any overhead crane system, at less cost and with greater flexibility and capacity. His plan was to arrange a structure having cableways with both transverse and longitudinal motion, and this by parallel track carriage motion in addition to the usual cable carriage motion. The Eastern Shipbuilding Company contracted for the construction of two large vessels for the Great Northern Steamship Company before the building of the New London plant had commenced, but after the plate shop was in successful operation, work was commenced on a wire rope cable system for handling material over the building berth, upon which these large steamships were to be constructed. These vessels, it may be of interest to note, are about 630 ft. long, 74 ft. beam, and 55 ft. deep to the strength deck.

This trolley installation is a broad departure from any method of handling material in engineering work adopted up to date, and the writer hopes for a successful future for such wire rope cable installations. The general arrangement of the New London cable system is shown on Plates XXXV., Fig. 3, and XXXVI., Figs. 1 and 2. There are three steel spars, each 120 ft. long, 42 in. maximum diameter, 30 in. diameter at the base, and 24 in. diameter at the head. Each steel mast supports a steel cross-yard 174 ft. long, of rectangular cross section, strongly braced. The middle yard is 4 ft. 6 in. deep and 6 ft. 3 in. wide in the centre, tapering to 3 ft. 2½ in. deep at the ends. The end yards are 4 ft. 6 in. deep and 5 ft. 2 in. wide in the centre, reducing to 2 ft. 6 in. deep and 2 ft. 9½ in. wide at the ends. The top of the yards are 84 ft. from the ground, and the end yards are well trussed by wire ropes, there being a compression strut 15 ft. long running from the centre of the back of each end yard away from the trolley field of operation. The three steel masts with their cross-yards are located midway between the two building berths, upon which now rest the 20,000 ton Great Northern steamships, the distance between the masts being 300 ft. The cross-yards have a length sufficient to cover the large beam of these vessels. The working field of the trollies is, therefore, a rectangle, approximately 600 ft. long and 174 ft. wide.

This steel construction is held in position entirely by the wire rope guys. Each yard is held up to its work by steel ropes running from the mast head. There are six of these lifts on the main yard, and four on each of the fore and mizzen yards. From the end of each yard, rope guys run to the ground, and these are securely fastened to granite anchorages buried in the ground. On the main yard, two guys run from each end to ground anchorages, one set of guys being, like the third pair of lifts on the yard, merely emergency or auxiliary members. From the centre and end of each yard, parallel wire rope guys connect the yards together, and diagonal guys connect the end of each yard to the centre of each mast next to it. These diagonal guys are practically only emergency members, but they also tend to give stiffness and rigidity to the structure. From the ends of the fore yard, guys run on a pronounced angle to solid granite anchorages, and from the centre of this yard two very heavy guys run to similar but heavier anchorages well out from the base of the mast. These members are of great importance, as they alone prevent the structure from falling aft. The guys from the ends of this yard are made large to reinforce the centre guys, which, as above stated, are in duplicate. At the water end, there is a heavy guy secured to a large concrete anchorage confined on the river bed by well timbered walls, which keeps the structure from falling forward.

From each end of this yard, a wire rope guy runs to the stone boundary wall of the yard, the base of these triangles being only 40 ft. On account of the undesirability of placing responsible anchorages under water, where metal corrodes, and vital members are practically inaccessible, a well-trussed steel "strongback" or compression member of rectangular box section has been fitted at an angle of 45 degrees, running from the mizzen mast just below the cross-yard forward to the ground, midway between the building berths. This strongback, or strut, has a maximum depth of 33 in., and a width of 26 in. It reduces at the ends to 27 in. deep, the width being constant. All the standing wire rope guys of the structure are galvanised steel cables, of the type used in America for suspension bridges. They have very little stretch, and are formed of six strands with wire centre, each strand, and the centre likewise, being made of seven wires. These ropes have an unusually high tensile strength, and are made with a view to long life when exposed, and they have great effective durability and rigidity.

The following table gives the weight and strength of the ropes used:—

Diameter. Inches.	Circumference. Inches.	Weight per Foot. Lbs.	Ultimate Strength. Tons.
$1\frac{3}{8}$	$4\frac{1}{4}$	3·10	75
$1\frac{1}{2}$	$4\frac{3}{4}$	3·70	90
$1\frac{5}{8}$	5	4·34	106
$1\frac{3}{4}$	$5\frac{1}{2}$	5·10	124
$1\frac{7}{8}$	$5\frac{7}{8}$	5·90	144
2	$6\frac{1}{4}$	6·73	164
$2\frac{1}{8}$	$6\frac{5}{8}$	7·60	185
$2\frac{1}{2}$	$7\frac{7}{8}$	10·50	256

Each rope guy is fitted with turnbuckles with sufficient play to allow for stretch and variation of temperature. Upon each side of the top of the main yard, and on the aft top side of the fore yard and forward top side of the mizzen yard, a track is formed of two continuous lines of Z bars facing each other and spaced 1 foot apart. A carriage composed of four wheels, suitably connected, runs horizontally in the track, bearing on the side of the Z bars nearest its field of work. Two such trucks, one on one side of the main yard, and one on either the fore or mizzen yards, are connected by means of a heavy 2 in. wire rope cable, upon which a self-oiling modified Roebling cableway carriage runs. This carriage consists of two wheels bearing on the cable rope, and two hauling rope sheaves below it, excluding the hook sheave, and also two fair-lead sheaves above the carriage wheels. A becket is placed on the lower hook sheave, and the hoisting rope leads over the after sheave, under the hook sheave, and over the forward guide sheave to the hoister drum. A neat framework holds the wheels and sheaves in position, and prevents the carriage from leaving the rope cableway. Another hauling part connects each side of the carriage to give the same fore and aft motion, and the return line to the drum is kept in position by an upper pair of fair-lead sheaves. This is arranged as an endless rope to another drum on the hoister. The trucks at the ends of the cableway are operated athwartships simultaneously by means of two endless small wire rope connections to a third double drum on the hoister.

The present installation at New London consists of two movable trolleys, but the structure is capable of operating continuously four trolleys, each loaded with 5,000 lbs. weight. The operating machinery is installed in a house on the main mast, located just below the cross yard. This house is rectangular in plan, and resembles somewhat the fighting top of a military mast. It is securely fastened to the mast, which runs through the centre of the house. There are at present located in this house two hoisting machines, each operating an independent trolley, one taking the forward portion of two vessels, and the other covering the after portion. Each machine is operated by a 35 H.P. continuous current motor, 120 ampère, 250 volt (General Electric, Form G, Class 58), having a speed of 550 revolutions per minute, geared so as to give 68 revolutions per minute for the hauling and hoisting drums. The speed of the hoister is such that it requires only 10 seconds to lower the block and hook or grabs from the cable to the ground. The heaviest load can be hoisted up the staging runway in 30 seconds. The trolley will run with its full load the whole length of the yard from side to side of the double building berth in 30 seconds, and in the same length of time the trolley carriage on the cable way rope will travel from mast to mast. As the entire structure is held in position by wire rope guys—tension members only, for all the connections to earth capable of withstanding compression are in a true plane midway between the two building berths—it is apparent that the anchorages to which the wire rope guys are secured have a tremendous responsibility imposed upon them. Indeed, it may be said that the stability of the whole structure depends upon the efficiency and reliability of the ground anchorages and the connections to them.

Appreciating this point, the factor of safety on all guys connecting to the ground has been increased from 5, the usual factor, to 8, and in some cases, as before stated, emergency guys have been added. The anchorages consist of granite rocks, laid one above the other at right angles and locked with smaller rock. The foundations for masts and strut consist of wood crib-work resting upon a solid granite ledge, which runs through the length of the building slips a few feet below the ground surface.

As already stated, this cable system has given entire satisfaction. The stationary guys and yards are used for erecting purposes, besides the movable cables. The system does rapid and positive work, is thoroughly reliable, and rapidly inspires confidence. The two trolleys, one for each ship, have proved capable of handling all the material that can be laid out by the fitters, and taken care of by the erecting and bolting up gangs, when placed in position. They have frequently handled upwards of 100 tons of steel plates and shapes per day. The stated safe average load for each of the four designed trolleys, viz., 5,000 lbs., is much lighter than the load usually assumed in the design of crane service over shipbuilding berths, but as all the framing, beams, bulkheads, and plating are generally erected as individual bars or plates, the designed load is seldom exceeded. The heaviest plate used on the Great Northern steamships is 6,000 lbs., and the average is nearer 2,500 to 3,000 lbs.

At the yard of the William R. Trigg Company, Richmond, Virginia, a wire rope cableway is at present being used over the shipbuilding berth now occupied by the United States cruiser *Galveston*. This cableway is only a temporary arrangement, and it will be replaced at some later date with a gantry crane, previously described, running upon wood trestles. The cableway now in use has a span of 360 ft., and is placed over the centre line of the cruiser. The cableway rope, which is 1½ in. in diameter, has no athwartship motion, and is secured at each end of the span to a built up tower, 80 ft. high above the ground at the building berth. Both the tail and head towers are securely guyed to the ground, the anchorage of each being 388 ft. out from the base of the tower. These main shore guys are in the same plane as the cableway. There are also two guys from each tower, which run diagonally inward some 65 ft., spreading at the same time about 70 ft. from the centre line. The designed load of the cableway is 5,000 lbs. The cableway is operated by a double tandem friction drum electric hoist, direct current, of Lidgerwood make, with a General Electric motor (640 revolutions, 500 volts) attached.

The hoist is geared 10·09 to 1, and has a rope speed of 350 ft. per minute. The rear drum carries the conveying rope, and the front drum the hoisting rope. This hoist is placed on the ground 62 ft. forward of the head tower. The cableway and hoister are shown on Plates XXXV., Fig. 4, and XXXVI., Fig. 3. It will be noticed that the *Galveston* is being built on a level keel, and will be launched sideways. A fixed cableway over the centre of a ship undergoing construction with a trolley carriage fitted to it, which gives only fore and aft travel, although of some use, is decidedly limited in its operation. Its great failing is its inability to take care of shell plating, and to place, where desired, deck and bulkhead plating, &c. It has, however, done good work, and when worked in conjunction with boom swing derricks, the advantages are said to have more than outweighed the small cost of this installation. The W. R. Trigg Co. have now under consideration the use of a similar outfit, only with a much longer span, viz., about 1,300 ft., to work over three smaller vessels placed end to end in line.

The Lidgerwood Company, of New York, who designed the cableway now in operation at the above plant, have also within the last few months designed a cableway system for the yard of the Townsend & Downey Company, Shooter's Island, New York, which however has not been built as yet. The arrangement planned is somewhat similar to that above described at the Trigg yard, but as the vessels launch stern first, the long tail shore guy has been omitted, and two shorter guys with compression struts forward have been substituted. The tail tower and its two compression struts at the after end of the cableway span resemble a tripod in arrangement. The forward head tower consists of two spars, arranged in a V-shape, with a long ground guy running forward to an anchorage in the same plane as the main cableway. The installation proposed consists of three cableway spans, each entirely independent of the other, and the three are supposed to feed two building berths.

Fig. 1 (Plate XXXVII.) shows the general arrangement of these wire rope cableways as arranged for the construction of two vessels each about 50 ft. beam. The location of the plate shop is the cause of the three cableways being of different lengths. The first, placed midway between the two ships, has a span of 425 ft., the cableway on the starboard side of the two vessels has a span of 514 ft., and the one on the port side of the two vessels has a span of 429 ft. A Lidgerwood cable carriage runs upon each cableway, giving the desired fore and aft motion. The capacity of each cableway is 8,000 lbs. At the base of each of the forward towers is located the steam hoisting engine with 9 in. by 10 in. cylinders, and 33 in. diameter tandem friction drums. This arrangement of overhead cable service, although an improvement over the system which allows for only one cableway to be suspended over the centre of each vessel, has, nevertheless, many serious disadvantages. Although it is claimed that the cableways are located so as to handle the side plates efficiently, and that two of the cableways operating jointly will handle such members of ship construction as deck beams, bulkheads, &c., yet it seems to the writer that the scheme adopted by the Eastern

Shipbuilding Company, which permits of both longitudinal and transverse travel, is absolutely essential to a successful cableway installation.

(8) Locomotive Cranes, Standard Type, on Tracks alongside Berths.

Locomotive cranes operating on ground tracks alongside shipbuilding berths cannot be considered as an overhead crane service, and such cranes, although very valuable for transporting and handling of material in general, are of very little use in erection, as their effective field is very limited. Swing boom derricks with steam winches are necessary, and universally used where an efficient overhead crane service has not been installed. Space does not permit of a description of the facilities for handling material in American yards that are equipped with only ground travelling cranes and swing derricks.

(9) Locomotive Cranes, Derrick Type, on Tracks alongside Berths.

An interesting method of handling material over shipbuilding berths is in successful operation at the yard of the Maryland Steel Company, Sparrows Point, Maryland. It consists of an installation of locomotive cranes, derrick type, running on tracks between the ships, there being one large derrick or hoist tower moving on a 16 ft. track between each set of building berths. The first locomotive derricks of this type, built by the Maryland Steel Company, were of 10,000 lbs. capacity, and were considered experimental. One of these is illustrated on Plate XXXVI., Fig. 4, supplying material to two large suction dredgers then building. The derrick post or tower is a well-trussed light structure rectangular in plan throughout its height; an ordinary swing boom is attached to this derrick post on each side, each boom being intended to feed one ship.

Experience gained with these derrick cranes suggested modifications which have been embodied into the design of the larger new locomotive derricks, now in successful operation at the above-mentioned plant. These cranes are illustrated on Plates XXXVI., Fig. 5, and XXXVII., Fig. 2. They consist of a well-trussed frame structure, 65 ft. high from the ground, into which is set a revolving derrick post 80 ft. long, 34 ft. of which is housed below the bearing on top of the outer frame support. The derrick post or mast revolves upon roller bearings, and at its lower extremity has a ball bearing step. Both the outer structure and the revolving derrick post are made as light as possible, and are of rectangular section throughout, being well trussed and braced. Upon each derrick post is secured a 64 ft. trussed boom, which is led to its field of work by the rotation of the derrick post, the bearings being such that the derrick mast and boom can make a complete revolution, if desired. The capacity of each of these derricks is 25,000 lbs.; they have a height of 110 ft.

under the normal position of the boom, and sufficient overhang to reach the centre of two vessels 110 ft. apart. The hoisting speed is 90 ft. per minute, and travelling speed 150 ft. per minute. These cranes travel on tracks set at a 1 per cent. gradient, which is well supported and braced by piling and timber cross ties. The rail bars are of ordinary steel railroad section, weighing 100 lbs. to the yard, and these are laid on heavy wooden stringers placed above the cross ties. The cranes are actuated by a vertical hoisting engine, having a sufficient number of drums and proper connections to effect the various movements of hoisting, track travel, raising, lowering and rotating of the boom.

The general methods of design and construction of the cranes are clearly shown in the appended Plates. The rotation is effected by means of wire ropes around a large drum at the bottom of the mast, the rope leading to proper arrangements on the engine. It is claimed that these derrick cranes have proved quite effective in handling material. One advantage, which should be mentioned in connection with the smaller 10,000 lb. cranes, is the fact that they can be moved from one track to another, so that two cranes may be used on one track, or they may be arranged in any manner to suit the work in hand. It should also be mentioned that the larger cranes above described have sufficient spread of track and sufficient height under the platform of the crane, so that an ordinary flat freight car can be run through underneath the crane, delivering material to the second one, if it is so desired. This central track is being used in every instance, and by means of it material can be delivered to the crane, so that the crane itself is used almost continuously for hoisting and placing material. Fig. 3 (Plate XXXVII.) shows the arrangement of building berths and crane service.

Concluding Remarks.

One of the features of crane service over shipbuilding berths that has promoted a good deal of discussion is the number of cranes or individual hoisting and conveying carriages required over each building berth. The Table on the next page gives the number of cranes per berth adopted in American yards, a representative installation of each type being taken as a basis for comparison.

A large number of points must be considered in determining the number of hoisting units required over a shipbuilding berth. Arrangement of shops, location of positions where material is landed for the cranes to pick up; track, trolley, and hoisting speeds, and size of vessels to be constructed, have all a great bearing upon this subject. The writer's experience seems to indicate that with a well-arranged lay-out, eliminating crane travel as much as possible, and with high speeds throughout, one hoisting and conveying unit will handle about as much material as can be fed to it, and taken care of when distributed over any merchant vessel up to about 7,500 tons register. Beyond this size it is probable that the desired rapidity of construction would warrant the installing of two crane units per vessel. If the cranes are operated both night and day, one hoisting and conveying unit will supply all the material desired for the construction of the largest merchant vessel built or projected.

Type of Crane Service.	Shipyard.	Extent of Building Berth covered by one Crane.
Cantilever	Wm. Cramp & Sons.	One crane for two vessels.
Gantry with single cantilever arm.	Craig S. B. Co.	One crane for each vessel. Propose to add another smaller crane.
Gantry with double cantilever arm.	Buffalo D. D. Co.	One crane for one vessel.
Ship House roofed over and enclosed on sides.	New York S. B. Co.	Two cranes for wide vessels. One crane for narrow vessels.
Ship House roofed over, open sides.	Fore River S. & E. B. Co.	Generally two cranes for each vessel.
Ship House frame structure ...	Union Iron Works.	Two cranes for smaller class of vessel. Four cranes for larger class of vessel.
Wire rope trolleys	Eastern S. B. Co.	One trolley for each vessel. (Could be increased to two with slight additional expense, if desired.)
Locomotive derrick cranes ...	Maryland Steel Co.	One for each vessel. When berths are not all occupied, two for one vessel sometimes.

The writer has refrained from criticising the various installations above described. Location and certain peculiar conditions existing at any plant at the time an overhead crane service is decided upon, exert much influence upon the type of service finally adopted. America is a large country, and the climate and atmospherical conditions vary throughout its extent. Materials and labour are likewise a variable and dictating quantity, and these features, added to the business policy of a shipbuilding company, make it extremely difficult to intelligently criticise in general any crane service installation. For a plant located in New England, where the air is clear and dry and the temperature normal, and especially at certain points where high winds are very common, the writer would not recommend a roofed-over building berth. Pneumatic tools are responsible for noise, and the roof with open sides creates draughts, and these drawbacks have a most pronounced detrimental effect upon the workmen. Moreover, lofty ship houses with open sides are a very inferior protection in stormy weather, and at times a small ship, building in a lofty ship house, becomes almost as wet and much more disagreeable to work on than would be the case if the vessel was building on an open berth. It is also almost impossible to obtain a good working light for ship construction with a roofed-in ship house. Skylights tend to remedy this evil, but their extent is limited, and no skylight occupying only, say, one-fourth to one-third of the roof area can ever give anywhere near as much light as would come from an open roof. There cannot be too much light for ship construction, and even when vessels are built in the open, there are places between deep vessels where want of light becomes objectionable.

The enclosed ship houses, that is, roofed over and side covered, overcome in general the annoying features of draught, but when large openings are made in the house sides, they are almost continually open, and these local draughts become at times even more disagreeable than the rush of air resulting from a complete open side. With the enclosed house, the absence of a good working light becomes more pronounced, and with some arrangements of shops and crane service, the getting of material to the ship and cranes is interfered with and seriously handicapped by the closed sides of the shed. The question of noise also becomes a serious factor, for pneumatic tools disturb the air with sound waves, which are reflected back to the ship, and produce a nerve-wearing, irritating, disagreeable clatter, with a rumbling and thundering which baffles description. It becomes utterly impossible for men to communicate with each other, except by signs.

A skeleton frame structure, with roof truss, but no covering over it, overcomes the disagreeable features before mentioned, but it has some peculiar disadvantages which would mar its efficiency unless the plant was arranged to suit its adoption. With the side posts spaced close together, as in the Union Iron Works sheds, the cranes cannot be fed from the broadside. The design of such a structure which permits the use of the side posts as staging uprights is, however, a good point. The gantry cranes, with short cantilever arms running upon wooden trestles, have this same feature without the roof truss, and have the additional advantage of being able to pick up material on the broadside or end, as desired. Moreover, if material is fed to the building berths on the broadside, two or more gantry cranes can be operated with advantage upon the same trestles. The range of adjustment in staging uprights with different beams of ships would be the same as with the ship house skeleton frame. Gantry cranes with double cantilever arms could be built where end launching is practised, so that material could be hoisted on either broadside clear of the staging. The gantry crane with short cantilever extension must of necessity be cheaper than the cantilever cranes, on account of the much greater distance between its supports and the shorter arms.

A cantilever crane, designed to hoist a heavy ship construction load at the extreme end of its arm, requires counterweighting, and even then becomes a heavy, expensive structure, with much greater average hoisting power than is usually required. The trestles of a cantilever crane, as usually installed, take up a good deal of room between building berths, the staging uprights being usually clear of the structural trestle work; but, as already stated, such a criticism does not of necessity make this feature a disadvantage, for several yards have ample room to bestow on such a construction. If material was always hoisted from the end of the ways, or from one or both the outer broadsides of a couple of building berths, then a trestle could be built, preferably of timber, the posts of which would form the staging uprights, and thus reduce the distance between building berths, and save somewhat in cost of staging. At the West Superior yard, it will be noticed that the steel trestle of the cantilever crane forms a permanent scaffolding.

The wire-rope cable system as perfected at the plant of the Eastern Shipbuilding Company requires broadside feeding of material. All the steel is fed to both ships from the starboard side of the vessel nearest to the plate shop. The overhead trolley system is a most excellent arrangement, requiring no trestle work, and the vessels can be placed as close together as the staging uprights permit. It has the possible objection of the ground guys interfering with adjoining berths, in case many berths are covered with this method of handling material. This disadvantage could be overcome, however, and for cost, room occupied, rapidity of operation and flexibility, this system seems almost ideal, provided the plant is laid out with a view to its adoption.

The locomotive derricks or hoist towers occupy about as much room as a cantilever crane, and the constant rotating of the jib makes their effective action slower. Moreover, three cranes are required to feed two vessels, in order that each side of each vessel can be covered.

The frequent visits of British and foreign naval architects and engineers to America, having for one of their objects the study of American methods of handling material in shipyards, has influenced the writer to prepare this extensive paper, hoping that it may prove of value, as well as of general interest to many of the members of this Institution.

APPENDIX I.

Sizes and Capacities of Locomotive Cranes furnished to American Shipbuilders by the Three Leading American Crane Manufacturers.

Maker.	Shipbuilding Co.	No. of Cranes.	Maximum.		Normal.		Wt. Mach. with counterweight.	Size of Engine.	Engine Revs.	Wheels. No. and Diameter.
			Arm Radius.	Hoist Weight.	Arm Rad.	Hoist Weight.				
			Ft.	Lbs.	Ft.	Lbs.	Lbs.	Double.		
*Brown ...	Wm. Cramp & Sons	6	26·5	9,500	14	22,000	74,000	9 × 7	350	4 @ 28″
„ ...	Newport News Co....	2	26·5	9,500	14	22,000	74,000	9 × 7	350	4 @ 28″
„ ...	New York S. B. Co.	1	26·5	9,500	14	22,000	74,000	9 × 7	350	4 @ 28″
„ ...	Fore River	1	26·5	9,500	14	22,000	74,000	9 × 7	350	4 @ 28″
„ ...	A. Sewall Co. ...	2	18	3,800	12	6,000	18,000	5½ × 9	266	4 @ 20″
„ ...	Risdon Iron Works	1	18	3,800	12	6,000	18,000	5½ × 9	266	4 @ 20″
„ ...	Risdon Iron Works	1	24	5,400	12	15,000	45,000	6½ × 10	240	4 @ 28″
„ ...	West Bay City ...	1	26·5	9,500	14	22,000	74,000	9 × 7	350	4 @ 28″
„ ...	Union, Buffalo ...	1	26·5	9,500	14	22,000	74,000	9 × 7	350	4 @ 28″
„ ...	Townsend & Downey	1	18	3,800	12	6,000	18,000	5½ × 9	266	4 @ 20″
†Industrial .	Chicago S. B. Co. ...	1	25	7,000	10	20,000	63,000	8 × 10	—	4 @ 24″
„	Detroit D. D. Co. ...	1	30	6,000	10	20,000	64,000	8 × 10	—	4 @ 24″
„	Cleveland S. B. Co....	1	30	8,000	11	20,000	70,000	8 × 10	—	4 @ 24″
„	West Superior ...	1	30	11,000	12	30,000	100,000	9 × 12	—	4 @ 33″
„	New York S. B. Co.	1	38	5,000	12	20,000	66,000	8 × 10	—	4 @ 24″
„	Eastern S. B. Co. ...	1	38	5,000	12	20,000	66,000	8 × 10	—	4 @ 24″
„	Eastern S. B. Co. ...	1	30	4,000	16	10,000	52,000	7 × 10	—	4 @ 24″
„	Harlan&Hollingsworth	1	30	4,000	16	10,000	50,000	7 × 10	—	4 @ 24″
„	Union Iron Works...	1	25	8,000	12	20,000	66,000	8 × 10	—	4 @ 28″
„	Risdon Iron Works	1	25	8,000	12	20,000	66,000	8 × 10	—	4 @ 28″
‡Williamson	Wm. Cramp & Sons	4	18	10,000	11	14,000	27,525	8 × 10	240	4 @ 28″
„	Maryland Steel Co.	1	18	10,000	11	14,000	27,525	8 × 10	240	4 @ 28″

* Brown = The Brown Hoisting and Conveying Machine Co., Cleveland, Ohio.
† Industrial = The Industrial Works, West Bay City, Michigan.
‡ Williamson = The Williamson Bros. Co., Engineers, Philadelphia, Pa.

APPENDIX II.

General Particulars of the Brown 10-Ton Locomotive Crane.

Length of boom	26 ft. 6 in.
Hoist at 14 ft. radius	22,000 lbs.
„ 20 „	13,800 lbs.
„ 26 ft. 6 in. radius	9,500 lbs.
Hoisting, full load	$42\frac{1}{2}$ ft. per minute.
„ empty hook	164 ft. „
Rotating, full load	4 complete turns per minute.
„ empty hook	6 „ „
Track travel, full load	500 ft. per minute.
„ empty hook	630 „ „
Possible grade, with full load	6 per cent.
„ „ empty hook	7 „
„ curve, with full load	70 ft. radius.
Wheels	4 of 28 in. diameter.
Axles in journals	5 in.
„ wheel seat	6 in.
Engines, pair vertical	9 in. diameter by 7 in. stroke.
„ revolutions	350 per minute.
Gauge of track, standard	4 ft. $8\frac{1}{2}$ in.
Boiler, diameter	54 in.
„ height	8 ft. $3\frac{1}{2}$ in.
„ tubes	98 of $2\frac{1}{2}$ in.
„ shell	$\frac{3}{8}$ in. open hearth steel.
„ heads	$\frac{3}{8}$ in. flange steel.
Working pressure	100 lbs.
Water tank capacity	200 gals.
Coal supply	650 lbs.
Extreme height, top of rail to top of stack	16 ft. $8\frac{3}{4}$ in.
Extreme width of crane	10 ft.
Wheel base	8 ft.

Total weight of crane, without counterweight, coal, or water = 58,000 lbs.
Total weight of crane, with counterweight, but no coal or water = 74,000 lbs.

APPENDIX III.

Industrial Locomotive Cranes.

The Industrial Works locomotive steam crane, of 10 tons stated capacity, is self-propelling at a speed of $4\frac{1}{2}$ miles per hour; will haul upon level tracks two to three loaded freight cars. When light, will ascend a gradient of $3\frac{1}{2}$ ft. in 100 ft.; can be adapted to ordinary curves; has a hoisting speed of about 25 ft. per minute, and with a single line will lift 10,000 lbs. at a speed of 50 ft. per minute. It has a rotating speed of $2\frac{3}{4}$ revolutions per minute, and is provided with radius varying connections.

The 5 ton crane, made by the same Company, is self-propelling at a speed of four miles per hour, and on a level track will haul usually two loaded freight cars. When light, will ascend a grade of 3 ft. in 100 ft.; can be adapted to ordinary curves; has a hoisting speed for maximum loads of 25 ft. per minute, and with single line will hoist 5,000 lbs. at a speed of 50 ft. per minute. It has a rotating speed of 2½ revolutions per minute, and is also provided with radius varying connections.

APPENDIX IV.

A.—LIST OF NAVAL VESSELS OF OVER 9,700 TONS DISPLACEMENT BUILDING IN THE UNITED STATES, JANUARY 1, 1902.

Name of Vessel.	Type.	Displacement. Tons.	Builders.
Virginia	Battleship	15,320	Newport News S. & D. D. Co.
Nebraska	„	15,320	Moran Bros.
Georgia	„	15,320	Bath Iron Works.
New Jersey	„	14,600	Fore River S. & E. B. Co.
Rhode Island	„	14,600	„ „ „
Pennsylvania	Armoured Cruiser	13,800	Wm. Cramp & Sons.
West Virginia ...	„ „	13,800	Newport News S. & D. D. Co.
California	„ „	13,400	Union Iron Works.
Colorado	„ „	13,400	Wm. Cramp & Sons.
Maryland	„ „	13,400	Newport News S. & D. D. Co.
South Dakota	„ „	13,400	Union Iron Works.
Ohio	Battleship	12,440	„ „ „
Maine	„	12,300	Wm. Cramp & Sons.
Missouri	„	12,230	Newport News S. & D. D. Co.
St. Louis	Armoured Cruiser	9,700	Neafie & Levy.
Milwaukee	„ „	9,700	Union Iron Works.
Charleston	„ „	9,700	Newport News S. & D. D. Co.

APPENDIX IV.—(*Continued.*)

B.—List of Merchant Vessels of over 8,500 Tons Register, Building in the United States, January 1, 1902.

Name of Vessel.	Owners.	Tonnage.	Builders.
——	Great Northern S. S. Co.	20,000	Eastern Shipbuilding Co.
——	" " "	20,000	" " "
Minnekakda	Atlantic Transport Co.	13,400	New York Shipbuilding Co.
Minnelora	" " "	13,400	" " " "
Finland	American Line	12,200	Wm. Cramp & Sons.
Kroonland	" "	12,200	" " "
Korea	Pacific Mail Co.	11,300	Newport News S. & D. D. Co.
Siberia	" " "	11,300	" " "
Shawmut	Boston S. S. Co.	10,485	Maryland Steel Co.
——	" "	10,485	" " "
Missouri	Atlantic Transport Co.	9,000	" " "
Maine	" " "	9,000	" " "
Massachusetts	" " "	9,000	New York Shipbuilding Co.
Mississippi	" " "	9,000	" " " "
Texan	Hawaiian Line	8,600	" " " "
Alaskan	" "	8,500	Union Iron Works.
Arizonian	" "	8,500	" " "

APPENDIX V.

A.—STATEMENT SHOWING THE NUMBER AND TONNAGE OF STEEL VESSELS BUILT IN THE UNITED STATES DURING THE PAST FOUR YEARS.

Year ending June 30.	Total Number of Vessels.	Aggregate Tonnage.
1898	64	62,325
1899	92	131,756
1900	92	197,125
1901	120	262,699

B.—STATEMENT SHOWING THE NUMBER AND TONNAGE OF VESSELS (WOOD AND STEEL) BUILT IN THE UNITED STATES DURING THE PAST FOUR YEARS.

Year ending June 30.	Total Number of Vessels.	Aggregate Tonnage.
1898	952	180,458
1899	1,273	300,038
1900	1,447	393,790
1901	1,580	483,489

DISCUSSION.

Mr. Archibald Denny (Member of Council): My Lord and Gentlemen, although Mr. Fairburn may be young, I think he has done very good work, and he is not afraid of expressing his opinion; in fact, he is not quite so afraid as I am. I have been over to America lately, and I have seen a good number of these outfits. I had better go back to the year 1882, when I first went to America; I then saw Cramp's Yard, Roach's Yard, and Harlan and Hollingsworth's. My impression is, although I am not quite sure, that there were practically no other yards of any importance except the Union Ironworks, and I may say that the yards then were not as good as our own here. My second visit was in 1899, when I landed at San Francisco, and saw at the Union Ironworks these wooden sheds, at least up to No. 5; No. 6 was then just being constructed. In considering the advantages of those open sheds, you must remember that they are in California, where the climate is delightful—a sort of perpetual spring where, I believe, they have only some 10 inches of rain during the whole year—and where no question of the rotting of the wood or anything of that sort need be considered. I then visited Newport News, and saw these huge gantries. I also visited Cramp's again, and this time I found the yard very much extended, and they were just taking in 10 acres more to make a new engine shop. I also saw at that time two or three of the yards on the Lakes—the Chicago Shipbuilding Company, the Loraine, and the Globe Ironworks. Now all these yards had adopted some species of hoisting arrangement, but of them all I was most taken with the idea of the Union Ironworks—two cranes over each berth—and I was confirmed in my idea that two cranes are necessary if you wish to develop any great speed in building, by the opinion of the Chicago Shipbuilding Company's manager and the manager of the Loraine Company. They both told me that the cranes were excellent, but that they had invariably to work overtime to get the material built into the ship. I went back last year and visited the Fore River Shipbuilding Company, and also the Camden Shipbuilding Company, Philadelphia. It was in August, and the manager of the Fore River Shipbuilding Company informed me that in July of the previous year they started in a green field, and they had already, when I was there, erected the larger half of their buildings—their engine works were very nearly completed. They had not started to build the large ship house, but, just to keep themselves going, they had taken in hand a cruiser, and with the plant they already possessed, they had got her plated in the month of August. I do not know when they started her, but it could not have been before the end of the previous year. The green field, or a good deal of it, was still in the yard; they had not got it all covered. I did not see the New London system with the wire rope, but I can't say it impresses me. The New York Shipbuilding Company at Camden certainly was a revelation. Two years before, there was nothing there at all; this time I found all these sheds completed, the engine works and yard in full swing; they had delivered their first steamer, and there were two 370 footers under one shed, both nearly plated and riveted, and other larger steamers commenced. To give you some idea of the enormous size of these sheds, when I went down with the manager on board of the steamers, I guessed their size at 250 feet, but as a matter of fact they were 370. Everything was absolutely dwarfed by the enormous size of the sheds. The boldness with which they made the installation of cranes, especially with regard to that 110 ton crane, was remarkable. Their wet basin was covered by a roof and was broad enough to hold two large steamers, but this crane spanned it entirely. They had built all these berths exactly the same breadth, so that they could take this enormous crane, run it up to the end of their slip, put it on to the transportation carriage, run it along the upper ends of the slips and put it on to any one of them in a very short time. Of all the yards I saw, I certainly think that the Camden Yard was the boldest in conception. Of course, when seeing these things, you have a regret that you do not own them, however much objection you might have to pay for them, and you have the feeling that you also would like to start, as the Americans have been doing, in a green field. In considering the position of our own yards here, I think we must not blame ourselves for not having got so many of these outfits as they have, because our yards have grown up and been developed slowly, whereas theirs have jumped up; they have had the benefit of all our experience, and they have been in the habit, on account of their extremely expensive labour, of endeavouring to minimise the cost of labour as far as possible. In loading and unloading ore and coal, and in transportation of all descriptions of material, they have been compelled to adopt apparatus which we have not yet been obliged to do to such an extent. There is also this to be said, that while this paper may make us feel a bit humble, I do not think that, up till now, the Americans can give us points in the question of the cost of building, whatever they may do later. Be that as it may, we must feel grateful to Mr. Fairburn for the enormous labour he has given himself in writing this paper for the information of this Institution.

Mr. John Scott, C.B., F.R.S. (Member of Council): My Lord, I do not know that I can speak with very much advantage on this paper, because, unlike my friends Professor Biles and Mr. Denny, I have not had the advantage of visiting these establishments except by proxy, but I must say that the ingenuity—of which I am aware, and which this paper has laid before us very clearly—which has been shown by our cousins across the Atlantic in treating this rather difficult subject is certainly very remarkable. It is a question, however, in my mind whether the capital expenditure involved will prove remunerative. In this country, as far as I know, we have only one establishment of anything like an analogous form, that of Messrs. Swan & Hunter, at Newcastle. It is difficult for me to speak on a point of this kind, and it is a point on which I cannot indulge in any public thoughts, but I should like to know very much whether those gentlemen, comparing their cost of working with their neighbours', and in relation to capital, have found it advantageous. Their form of construction differs slightly from those that have been laid before us by Mr. Fairburn, but, except in a case where the basis of wages is very much higher than we have in this country, I doubt extremely whether, *quâ* capital, the expenditure will be advantageous. I have myself had to treat the subject in a foreign country for friends as their consulting engineer, and I have examined those American systems in view of laying out rather a large dockyard at Hong Kong, to ascertain whether some of those systems could be advantageously adopted for a yard and engine factory which are going to cover 36 acres of ground. On the principle which I have laid down, I have up to the present moment come to the decision not to follow the American system, because labour in the Far East—in China—is infinitesimally cheap, and, from that point of view, I thought the capital had better be saved. At home here, as Mr. Denny has well said, old establishments are entirely tied up from the arrangement of their yards having grown to a large size over a great number of years. They are precluded, to a certain extent at least, from hurriedly adopting the American methods, and I think that it requires very considerable attention to the monetary result of any expenditure before determining whether it would be desirable for us to make any change in our present methods.

Mr. H. E. Camps (Member): My Lord and Gentlemen, on page 254 I notice that the writer refers to the squads that do the work in some of the shipbuilding yards. He refers to the work done by the fitters and taken care of by the erecting and bolting-up gangs. I should like to ask Professor Biles if that means that the work of preparing the plates is done on the same system as in the English, Scotch, and Irish yards, viz., by a squad who first lift the templates off the ship and mark the plates, another squad who prepare the plates in the shop, and a third squad who take them from the shop and hang them on the ship. Here it would almost seem that the templates are lifted and the plates prepared in the shop by one squad, and that the rest of the work is done by the cranes themselves. Of course, in dealing with a paper like this, one is apt to compare the method used in America with what would be possible in this country, and in going through it, I should say that the wire cable system is the only one illustrated in the paper which would be at all readily applicable to our yards here. It takes up considerably less room, and would require less alteration in the arrangement of the berths, than the other systems described by Mr. Fairburn. As to the question of covering in new shipbuilding berths with sheds, I agree with the writer of the paper, inasmuch as, unless the whole berth is covered in at the top and sides, it is rather a disadvantage than otherwise. If the sides are open, draughts are created, which make it very uncomfortable for the men to work; and if the top is uncovered, rain can get in just as easily as where there is no shed at all; but if a shed *is* to be there, why should it not be utilised both for protecting the men from the weather and for facilitating the handling of material? Messrs. Swan & Hunter's works at Wallsend do both these, and I know that during the last two or three years they have closed in one side at least, as well as the top. Another question is that of the lighting. The berth, being covered over, naturally takes a certain amount of light away from the men down below. Messrs. Swan & Hunter's works are all covered with glass, which is a very much better arrangement, in my opinion, than covering the berths with wood or iron, and so keeping out the greater part of the light which ought to reach the bottom of the berths. I notice one thing in the photographs: I do not know whether it is always so, but in our English and Scotch yards a feature that strikes the visitor in going round them is the dirty and generally untidy condition of the works. Now, although the Americans have the advantage of us in their facilities for handling materials over the berths, they have not disposed of that chaotic appearance which exists in most shipbuilding yards, due to plates and bars and scrap lying about; and the appearance of the trestles between the berths adds to this impression. No doubt that is only an impression. Then with regard to those trestles, I noticed in one or two of the photographs that they were all built of wood. I am afraid that this could not be adopted in English yards, because our workmen are not particularly careful; hot rivets, and similar objects, are in the habit of dropping where they are least expected and least wanted, sometimes with disastrous results. I am afraid that, if trestles of that sort were used in our English yards, insurance premiums would be going up at a much greater rate than shipbuilders would like. I think the thanks of the Institution are due to Mr. Fairburn for bringing this paper before us, and at any rate, if we cannot adopt the methods illustrated in the paper, it gives us at least an idea of what is going on on the other side of the Atlantic, and should prove to English shipbuilders that it is quite time they were looking to themselves, and bringing their own yards into a state of greater efficiency than they are at present.

Mr. C. D. Doxford (Member): My Lord, there is one point I should like to have a little further information upon. There are various speeds mentioned in the different yards. The greatest, I think, is in the New York Shipbuilding Yard, where a speed of 700 feet per minute is mentioned. I should like to know if Professor Biles could tell us whether that is ever realised in actual practice. It seems to me rather an alarming speed, and one that we Englishmen dare not attempt in our present state of knowledge, but if it is really done on the other side, then I do not know why we should not do it here.

Professor Biles (Member of Council): My Lord and Gentlemen, I do not know that I am commissioned to reply to the discussion, or that I, in any sense, hold a brief for Mr. Fairburn, but perhaps I may be allowed to say something on the subject. With respect to the method of working plates, the method that is being aimed at in America in the most progressive yards is to template everything by first making templates in the mould loft, and then working those templates in the shop by comparatively unskilled labourers centre-punching the marks through the hole in the templates, and passing that plate on to the punching man and from him to the head of the ship, where the gang there takes the plate, and, by means of the crane, puts it on to the ship. In other words, the operation of plating goes through the hands of several different kinds of workmen. The Americans have yet to demonstrate to their own satisfaction, and possibly to ours afterwards, that this method does facilitate construction more than our own. The object aimed at by all these constructions and systems is *output*, and, therefore, in any estimate which is made of the value of the plant and the interest on the capital invested, a very much increased output must be reckoned as probable from this plant. What that increased output is, I do not think anybody yet quite knows. The biggest plants that exist in the States have the misfortune at the present moment to be in the early stages—early not only in the sense that they have recently been erected, but that their whole organisation and system of working are being developed, and I think some of them have done exceedingly well in the rate at which they have turned out work; and when they get into good working order, I think we shall get some valuable information from them as to what their output is with their newly developed plant. Until we get that, nobody can say whether the investment of so much capital is justifiable or not. That is the view which I think we must take on that question at the present moment. I think, my Lord, it would be a good thing—and I have said it in other places—if the Admiralty could be induced to put down a plant very much like the New York Shipbuilding Company's plant. They have every opportunity of employing a plant of that kind with well organised and economical labour, and with a complete supply of work to keep such a large plant going. I think if they would do it, it would teach us a good many lessons in shipbuilding, and the country would save directly in the cost of the ships built in the dockyard. I suppose we can hardly hope or expect that they will do that, but I am sure it would be a good thing if they would. I do not know that anyone made any remarks with regard to the cable system, but, as developed at the Eastern Shipbuilding Company's works, with its three masts and six systems of trolleys, it is equivalent to two cranes over each ship, so that it is rather better in that way than the ordinary system, which has only one crane over each ship. In the New York Shipbuilding Company's Yard, it took two cranes running side by side to do the work that is done by the cable system on one side of the mast, because you can work the fore end of the ship independently of the after end. That seems to me to be one of the advantages of the cable system, and, of course, that advantage could be increased by increasing the number of masts. One of the disadvantages of it undoubtedly is the many anchorages that are required to maintain it in position. I do not quite know how they get over the difficulty of expansion in some of these long cables. I have seen the plant at work, and it is a little surprising to one who has been used to positive action in the way of hoisting gear, to see the cables moving about in a perfectly erratic fashion, wobbling sideways and in a fore and aft direction at high rates of speed with a load gyrating through the air. But they get the material there very quickly, and it is a very cheap plant to put up. I do not think anyone need fear that the wooden trestles that are put up will be burnt by hot rivets. The rivets are not any hotter in America than they are here, and we do not get our staging often burnt down by hot rivets. The reason they use trestle work very largely is, of course, that wood is, or has been, very much cheaper than steel. Mr. Doxford asked about these speeds. I have not actually measured the speeds, but they certainly look frightful. I was assured by a man whose word I am prepared to take, that they had run at 600 feet a minute in the New York Shipbuilding Company's Yard, and they look like it. I do not know why they should not; there is no necessity to linger on the road, and when once the material is hoisted, you want to get it lowered into position as quickly as possible. I do not think one need hesitate to adopt plans that will give that speed, and if they will not work successfully afterwards, one can slow down a little.

The PRESIDENT: I think, Gentlemen, we must give Professor Biles a hearty vote of thanks for the manner in which he has read this paper, and I presume you will authorise me or the Secretary to write to Mr. Fairburn to thank him for the paper he has written.

The following has been received from Mr. FAIRBURN in reply to the discussion on his paper:—

I am much indebted to Professor Biles for the very able manner in which he has read my paper, and for his kindness in explaining various portions of it and replying so completely to the questions raised by the discussion. There is practically little for me to say in reply to these remarks. I beg to assure Mr. John Scott that, on account of the high rate of wages prevailing in the United States, it is necessary to adopt all the labour-saving devices possible, and we have found by experience that a crane service over building berths is not only desirable, but absolutely essential, if economical construction is desired. Almost all the installations in American shipyards have proved paying investments, and I can, therefore, conservatively say that the capital expenditure involved in the use of an overhead crane service in our country has certainly proved remunerative. Mr. Camp's suggested drawbacks to the use of wood trestles for a crane service are outweighed by their many advantages. Wood is usually plentiful and cheap in America; moreover, it can be procured very readily, and the wood trestles are often used in conjunction with, or to take the place of, staging supports. I do not know of any trouble being experienced with fires in American yards. Such installations make no difference to insurance rates, and fires are just as apt to occur with the ordinary staging as with crane trestles. It is seldom that rivet forges are placed on the outboard staging of a vessel, and in America, forges are almost always inboard. Indeed, during fifteen years' experience in American shipyards, I do not remember a single fire taking place on the staging of any vessel undergoing construction. Regarding Mr. C. D. Doxford's question, pertaining to the speed of cranes, I would say that at New London, it is a common sight to see the cable carriage travel forward and aft so rapidly that the man holding on to the guide rope has great difficulty in running on the plated deck fast enough to keep his hold on the line. If such a man was travelling at only one-third the speed of a 100 yard sprinter, the crane travel would be 600 feet per minute. I have frequently timed cranes, which have travelled with a load a distance of 300 feet in thirty seconds from and to a position of absolute rest. The lack of skilled labour in the United States has made construction by the *universal* and *mould* methods essential. American shipbuilders have proved to their entire satisfaction that, for certain parts of the ship, the *universal* method of laying off is economical, and saves greatly in both cost and time. By the adoption of this method, we can lay off and mark our plates at a price below British piece-work rates, and the decks, bulkheads, beams, keel, frames, floors, longitudinals, casings, and portions of the shell are marked, punched, and ready for erection many weeks before the ships are ready to receive the same on the stocks. I have seen entire decks, 630 feet long and 73 feet wide, marked and completely machined before any deck beams were in position, and a great saving in time of construction has resulted from the extensive and judicious use of the *universal* method of laying off. It may be interesting to note that, when placed in position, the holes in the decks above referred to were practically perfect, and a fair deck, according to mould loft shape, was the inevitable and satisfactory result. I thank the members of the Institution for their kind reception of my paper. All the remarks made are much appreciated, and I can but regret that I have not had the occasion to reply at the time to criticisms concerning the American methods of handling material over shipbuilding ways.

A COMPARISON OF FIVE TYPES OF ENGINES WITH RESPECT TO THEIR INERTIA FORCES AND COUPLES, THEIR INCREASES IN WEIGHT DUE TO THE ADDITION OF BALANCE WEIGHTS AND THE VARIATIONS OF TURNING MOMENT ON THEIR CRANK SHAFTS.

By Professor W. E. Dalby, M.A., B.Sc., Associate.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 21, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

At the Spring Meeting of 1899, I had the honour of explaining to the Institution a simple semi-graphical method of making the detailed calculations required in the course of the design of a balanced engine, assuming the connecting rods to be infinitely long; and in my paper communicated to the Institution in the spring of 1901, this semi-graphical method was developed into an analytical method of including the effect of the connecting rod, and also of estimating the magnitudes of the unbalanced forces and couples in engines designed without regard to their dynamical balance. Both these papers deal with the more theoretical side of the subject, and the practical questions of how much heavier the moving parts of engines of the usual type become when masses are added to neutralise their unbalanced forces and couples, what change in weight is introduced when the conditions of balance are satisfied in the initial design so that no balance weights are necessary, what changes in the form of the crank-effort diagrams may be expected when the crank angles are set to suit the balancing conditions, were not considered. The object of this paper is to consider these questions in detail, and to illustrate the various points by means of comparative diagrams.

The only way of balancing a three-crank marine engine of the usual type is by the addition of balance weights, or bob weights, to the moving parts. In this case, therefore, balancing necessarily means the actual addition of considerable masses of material to the machinery, which have no other duty but that of producing forces equal and opposite to the unbalanced forces caused by the motion of the moving parts which are concerned in doing the proper work of the engine. It is well known to the members of this Institution that Messrs. Yarrow, Schlick, and Tweedy made a departure from existing practice when they began to build engines in which the moving parts concerned in doing the proper work of the engine were so arranged that they were in balance amongst themselves. In engines of this kind no part of the machinery merely turns round or reciprocates for the sake of the forces its motion causes on the frame. No such arrangement is possible, however, unless the engine has, at least, four cranks. This condition and the progressive increase in the power of marine engines have together determined the gradual introduction during the last ten years of the four-crank engine into the Navy and the Mercantile Marine. Yet that the possibilities of balancing the four-crank engine have not been generally recognised is shown by the fact that many engines of that type have been and are still being built with their cranks at right angles, even when absence of vibration is imperative. Four cranks at right angles is just the one particular arrangement of a four-crank engine which makes it impossible to effect balance without the addition of balance weights. A change in the crank angles, however, and a small change in the mass of the moving parts is all that is necessary to obtain an engine in which the moving parts are balanced amongst themselves; to change, in fact, a four-crank unbalanced engine into a four-crank balanced engine of the Yarrow, Schlick, and Tweedy type. These changes cannot be made in any arbitrary manner. The masses, crank angles, and centres of cylinders must be mutually adjusted to satisfy certain conditions. These conditions need not be discussed here, nor the methods of finding a solution of the equations expressing them, as these problems have already formed the subject of papers by Messrs. Yarrow, Schlick, Taylor, Macalpine, Macfarlane Gray, and others; my own papers already quoted being entirely devoted to this part of the subject. In the following comparisons, therefore, results only will be given. The comparisons are made between five engines of the same power arranged in typical ways. These different arrangements are examined; first, as regards the magnitudes of their unbalanced forces and couples; secondly, with respect to the increase of weight consequent upon the addition of balance weights to the crank shaft to balance the forces and couples in the vertical plane as completely as possible; thirdly, with regard to the variations of turning effort.

Types of Engines.—Each of the five engines is supposed to develop 1,750 I.H.P. at 90 revolutions per minute, triple-expansion.

The first arrangement considered is that of the ordinary three-cylinder marine engine, with cranks at 120°, and, in what follows, is referred to as the *Triple* Type.

The second type considered is a four-cylinder engine, in which the sequence of cranks and sequence of cylinders are arranged in the way adopted in H.M.S. *Terrible*, and is referred to throughout as the *Terrible* Type.

The third arrangement of cranks and cylinders is the same as used in H.M.S. *Powerful*, and is referred to as the *Powerful* Type.

In the fourth type, the cranks are mutually at right angles following one another, as in the *Powerful* type, but the sequence of cylinders is that of the fifth type, that is to say, the low-pressure forward cylinder and the low-pressure aft cylinder are placed so that the high-pressure and intermediate-pressure cylinders lie between them. Thus the sequence of cylinders is—

Low Pressure aft; Intermediate Pressure; High Pressure; Low Pressure forward.

This sequence of cylinders was used for the first time, I believe, in the engines built for the S.S. *Innisfallen* by Messrs. Wigham-Richardson & Co., to an order received in November, 1895. The crank angles used in these engines were none of them 90°, as in the arrangement under consideration, and for this reason the type has been named the *Modified Innisfallen* Type. This type has been included because the sequence of cylinders above, with cranks at right angles, has come into general use in recent years both in this country and in America.

An engine of the Yarrow-Schlick-Tweedy type is the fifth arrangement considered, and is referred to throughout the paper as the *Balanced* Type. A study of the curves in the paper will show that this type possesses all the advantages of a balanced engine without any disadvantage as regards turning moment, and has the great practical advantage that the balancing is effected with only a slight change in the weight of the moving parts. The sequence of cylinders is the same as in the *Modified Innisfallen* type.

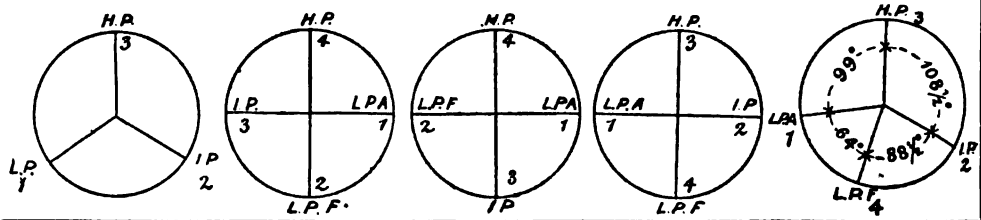
It should be distinctly understood that the weights and dimensions of the different types have no connection with or relation to the actual engines of the same name. The names have simply been chosen for convenience of reference because the arrangement of cranks and sequence of cylinders corresponding to them are so well known.

WEIGHTS OF THE MOVING PARTS.—These are shown in Schedule I. It will be observed that the weights are the same in gross and in detail for the *Powerful*, *Terrible*, and *Modified Innisfallen* types. The high-pressure and intermediate gear of the triple type is the same as in the above four-crank types, with the exception of small differences in the valve gear. The weight of the low-pressure gear is, of course, considerably different from the low-pressure gears of the other types. There are slight differences in the main gear of the balanced type due to the fact that the masses have to be proportioned to satisfy the balancing conditions. The valve gear in all the four-crank types has the same weight in detail and in gross. It will be observed that the total mass of the balanced engine compared with the other four-crank types is about 3½ per cent. greater. For the arrangement of the weights of the parts to fulfil the practical requirements of marine-engine design, I am indebted to my friend, Mr. Tweedy.

SCHEDULE I.

Detailed Masses of Moving Parts.

Stroke, 33 inches. Ratio of Crank to Connecting Rod, 1 : 4·44.
All masses are reduced to the crank radius of $16\frac{1}{2}$ in. The figures against each crank indicate their order, reckoning from the left hand.



Crank Number.		Gear.	*Triple* Type.	*Terrible* Type.	*Powerful* Type.	*Inniskfallen* Type.	*Balanced Type.* Yarrow-Schlick-Tweedy.
			Tons.	Tons.	Tons.	Tons.	Tons.
1	Recip.	Main	2·2250	1·1700	1·1700	1·1700	1·3370
		Valve	0·3024	0·2404	0·2404	0·2404	0·2404
	Revol.	Main	0·9000	0·6750	0·6750	0·6750	0·7410
		Valve	0·1428	0·1426	0·1426	0·1426	0·1426
2	Recip.	Main	1·6180	1·1700	1·1700	1·6180	1·6180
		Valve	0·2393	0·2404	0·2404	0·2404	0·2404
	Revol.	Main	0·9000	0·6750	0·6750	0·9000	0·9000
		Valve	0·1428	0·1426	0·1426	0·1426	0·1426
3	Recip.	Main	1·3500	1·6180	1·6180	1·3500	1·3960
		Valve	0·1768	0·2404	0·2404	0·1768	0·1768
	Revol.	Main	0·9000	0·9000	0·9000	0·9000	0·9000
		Valve	0·1428	0·1426	0·1426	0·1426	0·1426
4	Recip.	Main	—	1·3500	1·3500	1·1700	1·1700
		Valve	—	0·1768	0·1768	0·2404	0·2404
	Revol.	Main	—	0·9000	0·9000	0·6750	0·7400
		Valve	—	0·1426	0·1426	0·1426	0·1426
Total ...			9·0399	9·9264	9·9264	9·9264	10·2704

Comparison as Regards the Unbalanced Forces and Couples.—The series of diagrams given in the groups of figures A to E (Plate XXXVIII.) enables this comparison to be easily made, because all the curves are drawn to the same scale. In each group there are four separate sets of curves, the thick, unbroken curves giving severally,

The unbalanced force in a vertical plane;
The unbalanced couple in a vertical plane;
The unbalanced force in a horizontal plane;
The unbalanced couple in a horizontal plane.

In every one of the twenty diagrams, the angle between the several high-pressure cranks and the vertical plane is indicated horizontally. Thus, considering Group A, when the engine, the parts of which are detailed in Schedule I. under the head of "*Triple* Type," is running at 90 revolutions per minute, and the high-pressure crank is at 90° with the vertical plane, the instantaneous value of the

Unbalanced vertical force is	...	...	...	+ 1·7 tons weight;
Unbalanced vertical couple	...	...	...	− 50 foot tons;
Unbalanced horizontal force	...	...	...	− 0·3 tons weight;
Unbalanced horizontal couple	...	...	...	− 40 foot tons.

Or again, considering the curves belonging to the *Modified Innisfallen* type (Group C), under the same conditions of speed, the instantaneous values, when the high-pressure crank is 90° from the vertical position, are—

Unbalanced vertical force ...	...	...	...	− 3·2 tons weight;
Unbalanced vertical couple	...	...	...	− 54 foot tons;
Unbalanced horizontal force	...	...	...	− 1·5 tons weight;
Unbalanced horizontal couple	...	...	...	− 4 foot tons.

It should be understood that the curves showing the unbalanced forces and couples in the vertical plane give the effect of the reciprocating parts of the main gear, including the obliquity of the connecting rod, the effect of the reciprocating parts of the valve gear, assuming an infinitely long valve rod, and the effect of the revolving parts of the main gear and valve gear. The curves giving the unbalanced horizontal forces and couples include the revolving parts of the main gear and the valve gear. The curves for the vertical plane are, therefore, not simple sine curves like those for the horizontal plane, since they are distorted by the effect of the obliquity of the connecting rod.

In order that some idea may be formed of the relative effect of the main gear and valve gear in producing the unbalanced state, the maximum ordinate corresponding to the harmonic curve representing the effect of the valve gear is added to each diagram and marked "V," and the maximum ordinate of the simple harmonic curve representing the primary effect of the main gear is added and marked "P." In the horizontal plane the sum of the two curves determined by these ordinates results in the thick unbroken curve given in the diagrams. In the vertical plane the sum of the similar pair of curves differs from the thick unbroken curve given by the amount of the secondary effect due to the obliquity of the connecting rods.

Comparing the diagrams of Groups A and B (Plate XXXVIII.), it will be seen that no advantage is gained by an engine of the *Terrible* type over an engine of equal power of the *Triple* type. In fact, the unbalanced forces and couples are much greater in the former than in the latter type. Comparing the *Triple* with the *Powerful* type (Group D), there is a distinct advantage in favour of the four-crank engine. The *Modified Innisfallen* type (Group C) has an advantage over all the types represented in Groups A, B, D, as regards the vertical couple. Comparing any one of these types with the *Balanced Engine* of the Yarrow, Schlick, and Tweedy type (Group E), the advantage of this latter kind of engine is at once apparent.

THE BALANCING OF THE TRIPLE, "TERRIBLE," "POWERFUL," AND "MODIFIED INNISFALLEN" TYPES.—COMPARISON OF INCREASED WEIGHTS.—The question now arises, to what extent is it possible to reduce these unbalanced forces and couples by the addition of balance weights? It is only possible to completely balance the revolving masses, but the primary part of the unbalanced effect of the reciprocating masses in the vertical plane may be balanced by revolving masses attached to the crank shaft also, though these masses of necessity introduce forces and couples in the horizontal plane equal to those which they balance in the vertical plane. The secondary unbalanced forces and couples may, it is true, be balanced by suitably attached masses, but they must be operated twice as fast as the engine crank shaft.

This method of dealing with the matter was pointed out by Mr. Macalpine in an article in *Engineering* (February 19, 1897), and methods were given and illustrated of attaching masses to an engine to neutralise the secondary and tertiary effects. Confining myself to the balancing of the primary effect only, by the addition of balance weights to the crank shaft, these have been found for completely balancing the force and couple in the vertical plane. That is to say, when these weights are added to the crank shaft, the unbalanced force and couple in the vertical plane will be of a secondary order, and due to the obliquity of the connecting rod. Schedule II. gives particulars of these weights. It will be noticed that, to balance to this extent, requires the addition of mass to the moving parts of 52·2 per cent. of their original weight in the case of the *Triple* type, and to a less extent in the three other unbalanced four-crank type. The *Modified Innisfallen* type requires the smallest addition, namely 14·9 per cent. The unbalanced forces and couples remaining after the addition of these balanced weights are shown by the dotted curves on the diagrams of Plate XXXVIII. It will be noticed that in the vertical plane, the unbalanced force and couple curves are all of the second order. The effect in the horizontal plane is in most cases increased. In order to facilitate the comparison, the maximum values of the unbalanced vertical and horizontal forces and couples have been tabulated in Schedule III., both before and after the addition of the balance weights.

Comparison as Regards the Variation of Turning Effort.—If the work done were equally divided between the cylinders of the four-crank engines in which the cranks are at right angles, a good crank-effort curve would be obtained independently of the sequence of the cranks. Engineers generally prefer to arrange the distribution so that the work done in the two low-pressure cylinders combined is about equal to that done in the high-pressure or in the intermediate-pressure cylinders. In this case, the sequence of cranks becomes an important consideration. To obtain some basis of comparison in this respect, the starboard set of indicator cards (Figs. 5 to 8, Plate XXXIX.), taken at a full-power trial of H.M.S. *Powerful*, and published in *Engineering* (October 23, 1896), have been used in the construction of the crank-effort curves of the four types of four-crank engines under consideration. This set of diagrams is taken to represent the division of work between the cylinders most likely to be used in first-class practice, at the present time, for four-cylinder triple-expansion engines. This set of diagrams is used to construct a crank-effort curve for the *Balanced* type only for the sake of comparison. A crank-effort curve more appropriate to that type of engine has been constructed as well, from a set of diagrams taken from an actual engine of that type. These diagrams are shown in Figs. 9 to 12 (Plate XXXIX.).

The set of crank-effort curves, all of which have been corrected for the inertia of the reciprocating parts, are shown in Figs. 1 to 4 (Plate XXXIX.). It will be observed that the curve for the *Terrible* type (Fig. 1) is of very good form, the ratio of the maximum to the minimum torque being 1·68. The minimum torque is 0·73 of the mean and the maximum is 1·23 of the mean. The crank-effort curve of the *Modified Innisfallen* type would be precisely the same in form as this, if the component crank-effort curves belonging respectively to the low-pressure forward and low-pressure aft cylinders were identical in form. There is such a slight difference, however, that the one curve practically represents the turning moment of the two types. The crank-effort curve of the *Powerful* type is shown in Fig. 2. The variation of crank effort is now very much increased. An inspection of the component curves will show that this is due to the relatively small proportion of work done in each low-pressure cylinder. Fig. 3 shows the curve of the *Balanced* type constructed from the *Powerful* diagrams. It will be noticed that the variations of turning effort are not greater than the variation in the *Terrible* type, although the general shape of the curve is different. A great improvement in this curve is made by using the set of diagrams shown in Figs. 9 to 12 (Plate XXXIX.), which were taken from an engine of this type. The crank-effort curve from these diagrams is shown in Fig. 4. The ratio of the maximum to the minimum torque is now reduced to 1·33. The various ratios are set forth in Schedule IV.

In the construction of these curves, the following data were taken :—

High pressure		$25\frac{1}{2}$ in. diameter.
Intermediate pressure		$39\frac{1}{2}$ in. diameter.
Low pressure	each	43 in. diameter.
Stroke		33 in.
Ratio of crank to connecting rod, 1 : 4·44.		
Speed		90 revolutions per minute.
Mass of reciprocating parts, those given in Schedule I.		

The crank-effort curve corresponding to each cylinder is shown in the diagrams.

The Effect of Changing the Conditions of Comparison.—Any one of the five typical engines may be increased in power by increasing the speed. In this case, the curves giving the unbalanced forces and couples (Groups A to E, Plate XXXVIII.) remain the same in form, but they must be interpreted to a different scale. Thus, if the speed is changed from 90 revolutions per minute to n revolutions per minute, the space on the paper representing 1 ton weight will at the new speed represent

$$\left(\frac{n}{90}\right)^2 \text{ tons weight.}$$

Similarly, the distance representing 1 foot ton will now represent

$$\left(\frac{n}{90}\right)^2 \text{ foot tons.}$$

The change of speed will modify the form of the crank-effort curves (Figs. 1 to 4, Plate XXXIX.) to some extent; but, if moderate, will not destroy their general character, because, when the cranks are in the regions of 90° and 270°, the inertia correction for the reciprocating parts vanishes; and, at the dead centres, where the inertia effect is greatest, the turning moment vanishes.

The engines may be changed in power by changing their size, the speed remaining the same. Providing the weights are all changed in the same ratio, q, say, and the centres of the cylinders are all changed in the ratio p, say, the curves (Groups A to E, Plate XXXVIII.) remain the same in form, but the distance on the paper representing 1 ton weight will now represent q tons weight, and the distance on the paper representing 1 foot ton will now represent pq foot tons. If the change of weight of the reciprocating parts is made so that the weight per square inch of cylinder area is the same, the crank-effort diagrams will remain the same in form for the same set of diagrams for the same speed. If the weight per square inch of cylinder is not constant, there will be a change in the form of the crank-effort curves, though, as in the case of the change in speed, at the dead centres and in the regions of 90° and 270°, the crank-effort curve is unaffected by any change in the weight of the reciprocating parts.

If there is a change in size and speed, the change in size being proportionate, then the form of the curves (Groups A to E, Plate XXXVIII.) remains unchanged, though the distance on the paper which represents 1 ton weight will now represent

$$q\left(\frac{n}{90}\right)^2 \text{ tons weight,}$$

and the distance representing 1 foot ton will represent

$$pq\left(\frac{n}{90}\right)^2 \text{ foot tons.}$$

CONCLUSIONS.—The curves in Groups A to E (Plate XXXVIII.) and Figs. 1 to 4 (Plate XXXIX.) show that amongst four-crank engines with cranks at right angles, the *Modified Innisfallen* type has advantages over both the *Terrible* and *Powerful* types. In fact, it combines to some extent the advantages of both types without their disadvantages. Its unbalanced vertical couple is about the same as in the *Powerful* type, and considerably less than in the *Terrible* type. Its crank-effort curve is practically the same as in the *Terrible* type, and much more uniform than in the case of the *Powerful* type. The *Balanced* engine of the Yarrow, Schlick, and Tweedy type stands alone in the possession of the great advantage that the unbalanced forces and couples, both in the vertical and horizontal planes, are practically *nil*, and that this result is obtained without appreciably increasing the weight of the engine beyond the weight usual in contemporary first-class practice. The greatest unbalanced effect is a couple of the second order. The crank-effort curve from one of these engines can be made as uniform as the curves from any of the other four-crank types. Many four-crank, quadruple expansion engines of the *Balanced* type have been built, and there is no difficulty in securing a good crank-effort curve from them. The division of work between the four cylinders may be primarily fixed so as to satisfy the conditions of uniform crank-effort laid down in Dr. Lorenz's paper. As these conditions may be satisfied in many ways, there is ample scope to effect the division of work so that it satisfies the conditions imposed by practical necessity.

My thanks are due to Mr. Roberts, senior student in the Mechanical Engineering Department of the Technical College, Finsbury, for much help in working out the detail calculations, and in the drawing of the curves.

SCHEDULE II.

Details of the Balance Weights which must be added to the Moving Parts to completely Balance the Primary Forces and Couples in the Vertical Plane.

All masses reduced to the crank radius of 16½ inches.

The subscript figures give the angle between the radius of the balance weight and the high-pressure crank, measured in a counterclockwise direction.

—	*Triple* Type.	*Terrible* Type.	*Powerful* Type.	*Innisfallen* Type.	*Balanced* Type. Yarrow Schlick-Tweedy.
	Tons.	Tons.	Tons.	Tons.	Tons.
Balance weight attached to the crank shaft in the plane of the forward crank	$2{\cdot}2500_{148^\circ}$	$2{\cdot}7150_{220^\circ}$	$0{\cdot}8300_{220^\circ}$	$0{\cdot}5260_{108^\circ}$	Nil.
Balance weight attached to the crank shaft in the plane of the after crank	$2{\cdot}4700_{324^\circ}$	$1{\cdot}4950_{21^\circ}$	$1{\cdot}1100_{31^\circ}$	$0{\cdot}9530_{150^\circ}$	Nil.
Total added mass at crank radius	4·7200	4·2100	1·9400	1·4790	Nil.
Mass of unbalanced moving parts, from Schedule I. ...	9·0399	9·9264	9·9264	9·9264	10·2704
Mass of moving parts plus the mass of the balanced weights	13·7599	14·1364	11·8664	11·4054	10·2704
Percentage increase of mass due to the addition of balance weights	52·2	42·2	19·5	14·9	Nil.

N.B.—The difference between the total mass of the moving parts of the *Balanced* engine of the Yarrow-Schlick-Tweedy type, and the total mass of the moving parts of the engines of the *Terrible*, *Powerful*, and *Innisfallen* type, unbalanced, is 0·3440 tons, being a difference of 3½ per cent.

SCHEDULE III.

Approximate Maximum Magnitudes of the Unbalanced Vertical and Horizontal Forces and Couples before and after the addition of the Balance Weights given in Schedule II. for a Speed of 90 Revolutions per Minute.

Magnitudes before the addition of balance weights.

—	*Triple* Type.	*Terrible* Type.	*Powerful* Type.	*Inniskillen* Type.	*Balanced* Type. Yarrow-Schlick-Tweedy.
Max. Vertical Force ... *tons*	1·80	5·40	1·80	5·00	0·18
„ „ Couple ...*ft. tons*	120·0	200·0	83·0	52·0	23·00
Max. Horizontal Force ... *tons*	0·60	2·00	0·05	2·00	0·33
„ „ Couple *ft. tons*	45·0	75·0	27·0	17·0	3·30

Magnitudes after the addition of balance weights to completely balance the primary unbalanced forces and couples in the vertical plane.

Max. Vertical Force ... *tons*	0·70	0·23	0·60	0·23	—
„ „ Couple ...*ft. tons*	14·0	11·0	33·0	23·5	—
Max. Horizontal Force ... *tons*	1·75	3·80	1·30	3·00	—
„ „ Couple *ft. tons*	70·0	120·0	35·0	17·0	—

SCHEDULE IV.

Comparisons of Variation of Turning Effort.

—	*Terrible* Type.	*Powerful* Type.	*Innisfallen* Type.	*Balanced* Type.	
Mean Torque. 1,750 I.H.P. at 90 revolutions per minute. *ft. tons*	46·5	46·5	46·5	46·5	
				A.	B.
Maximum ÷ Mean	1·23	1·3	1·23	1·20	1·20
Minimum ÷ Mean	0·73	0·50	0·73	0·70	0·90
Maximum ÷ Minimum	1·68	2·60	1·68	1·70	1·33

The starboard set of indicator cards, taken from H.M.S. *Powerful* during a full-power trial (Figs. 5, 6, 7, 8, Plate XXXIX.), and published in *Engineering* (October 23, 1896, page 532), have been used in the drawing of the crank-effort curves, from which the above figures for the *Terrible*, *Powerful*, *Modified Innisfallen*, and the *Balanced* Type (Column A) have been obtained. The curve from which the figures in Column B of the *Balanced* Type have been obtained, was drawn from a set of cards (Figs. 9, 10, 11, 12, Plate XXXIX.) taken from an engine of that type.

DISCUSSION.

Mr. Macfarlane Gray (Member of Council): My Lord and Gentlemen, as I have devoted much attention to this subject, I rise to speak lest a wrong impression should be formed regarding my silence. I think Professor Dalby has done excellent work on this subject. I have avoided reading this paper critically, as I wish to have my mind relieved now from such studies. I do not think there is anything to criticise in it, for I believe it is all right, and I am sure we are all much indebted to the author for writing this paper.

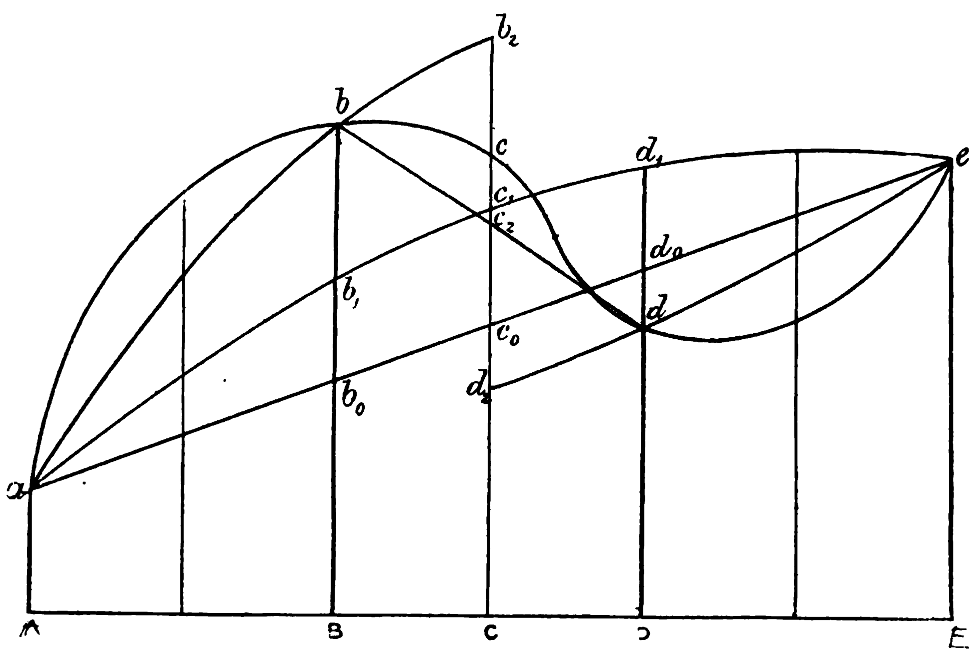
The President (the Right Hon. the Earl of Glasgow, G.C.M.G.): I am sure, Gentlemen, that you will all desire to give Professor Dalby your very sincere thanks for the interesting paper which he has given us this evening.

A NOTE ON SIMPSON'S RULES.

By J. Macfarlane Gray, Esq., Member of Council.

[Read at the Spring Meetings of the Forty-third Session of the Institution of Naval Architects, March 21, 1902; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

Simpson's Rules have two sets of multipliers, the 1, 4, 1 set and the 1, 3, 3, 1 set. The 1, 4, 1 set is based upon the common parabola, and the 1, 3, 3, 1 set is generally stated to be based upon a parabola of a higher order. This is misleading, as they are all founded upon the common parabola, and this Note is written to show how the 1, 3, 3, 1 set is obtained from the 1, 4, 1 set. Any parabola of the third order is merely a parabola of the second order with every ordinate increased by a constant multiple of the cube of its distance from the middle ordinate. As, at equal distances, these additions are



equal in amount but opposite in sign, the area remains unaltered, that of a parabola of the second order having the same middle ordinate, for which integration gives the multipliers 1, 4, 1.

The common parabola is a curve whose deflection from a chord at any point is proportional to the product of the segments of the chord at that point. The curves dealt with in naval architecture are assumed to be parabolas sheared perpendicularly to a base, the normal ordinates retaining their parabolic lengths, with the chord inclined to the base. In the accompanying figure the area to be measured is A a b c d e E, having an ogee curve for its upper boundary. According to the 1, 4, 1 rule the area is—

$$\frac{\text{A E}}{2} \cdot \frac{1}{3}(\text{A}\,a + 4\,\text{C}\,c + \text{E}\,e).$$

The two extreme ordinates and the middle ordinate are here employed, with the length A E. This gives the correct area up to the parabola which would pass through the points a c e, obviously much too great for the area required. Simpson no doubt saw that it would give a nearer approximation to substitute half parabolas on the same chord a e, through a b to b_2, and through e d to d_2, and to take the point c_1 midway between b_2 and d_2 as that through which the equivalent common parabola a c_1 e should be drawn. Prosecuting this idea, given the two intermediate ordinates B b and D d, he had to express C c_1 in terms of B b and D d in the 1, 4, 1 rule.

The chord a e is divided into six equal parts and the parabolic deflections at b_0, c_0, d_0 are therefore proportional to $2 \times 4 = 8$, $3 \times 3 = 9$, and $4 \times 2 = 8$, therefore—

$$c_0\,b_2 = \frac{9}{8}\,b_0\,b, \text{ and } c_0\,d_2 = \frac{9}{8}\,d_0\,d.$$

This amounts to saying that c_0 c_1 is $\frac{9}{8}$ times c_0 c_2, the mean of the deflections at b and d obtained by drawing a straight line from b to d. It is obvious then from the diagram, that—

$$\text{C}\,c_1 = \frac{\text{A}\,a + \text{E}\,e}{2} + \frac{9}{8}\left(\frac{\text{B}\,b + \text{D}\,d}{2} - \frac{\text{A}\,a + \text{E}\,e}{2}\right).$$

Now apply the 1, 4, 1 rule with this value for the middle ordinate—

$$\frac{\text{A E}}{2} \cdot \frac{1}{3}\left\{\text{A}\,a + 4\left(\frac{\text{A}\,a + \text{E}\,e}{2} + \frac{9}{8}\left(\frac{\text{B}\,b + \text{D}\,d}{2} - \frac{\text{A}\,a + \text{E}\,e}{2}\right)\right) + \text{E}\,e\right\}$$

$$= \text{A E}\left(\frac{\text{A}\,a}{6} + \frac{\text{A}\,a}{3} + \frac{\text{E}\,e}{3} + \frac{3}{8}\text{B}\,b + \frac{3}{8}\text{D}\,d - \frac{3}{8}\text{A}\,a - \frac{3}{8}\text{E}\,e + \frac{\text{E}\,e}{6}\right)$$

$$= \frac{\text{A E}}{3} \cdot \frac{3}{8}(\text{A}\,a + 3\,\text{B}\,b + 3\,\text{D}\,d + \text{E}\,e),$$

and this is the 1, 3, 3, 1 set of multipliers, or Simpson's Second Rule.

I apologise for taking up the time of the meeting with such a trifle, but these rules are of great importance, and the clearer our views are of the nature of the multipliers the more valuable they become.

DISCUSSION.

Mr. F. K. BARNES (Vice-President): My Lord, I have only a word or two to say at this late hour. I do not think this matter is really of vital importance, and it would be better, I think, to wait for Professor Bryan's contribution on the subject before we fully discuss it. I might say, however, that I do not think Mr. Macfarlane Gray has proved his case, namely, that of the parabola of the second order as against that of the third order, as stated in the first paragraph of the paper. He has not, at least, satisfied me.

The PRESIDENT: I am sure, Gentlemen, you will all join with me in thanking Mr. Macfarlane Gray for the paper he has brought before us. That, Gentlemen, closes the list of papers.

The following communication has been received from Professor G. H. BRYAN, F.R.S. :—

I had occasion to examine the methods of approximate integration associated with the name of "Simpson's rules" shortly before seeing the proof of Mr. Macfarlane Gray's paper, and, as I did not find the rule given in elementary text-books as "Simpson's rule" quite satisfactory for the purpose I had in view, it appeared easier to work the matter out independently than to consult more technical treatises, which are difficult to obtain in Bangor. The following remarks refer chiefly to the cases of approximate integration, where not merely three or four, but any number of equidistant ordinates are measured. The results, doubtless, are more or less well known, but a discussion of them cannot fail to be of interest, if only by attracting attention to the matter, and it may be convenient, for the sake of completeness, to work *ab initio*.

(1) *Trapezoidal Element of Area.*—If only the two end ordinates A a, B b of the area A a b B are measured, and if these be y_0 and y_1, their distance apart being h; then, if we have no data to determine

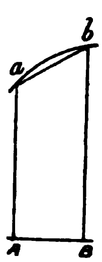


FIG. 1.

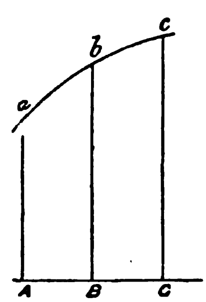


FIG. 2.

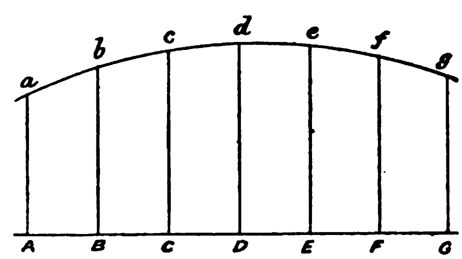


FIG. 3.

the shape of the arc a b, we cannot get a better estimate of the area than that obtained by treating the arc a b as a straight line, giving the well-known formulæ—

$$\text{Area} = h\left(\frac{y_0 + y_1}{2}\right)$$

Parabolic Element.—Next, supposing that three equidistant ordinates y_0, y_1, y_2 are measured, represented by A a, B b, C c, and let h be, as before, their distance apart (A B), then, as good an estimate as we can get of the shape of the curve will be that obtained by supposing the co-ordinates of *a*, *b*, *c* to satisfy an equation of the form—

$$y = A + Bx + Cx^2,$$

which equation represents a parabola with its axis parallel to the axis of y, *i.e.*, vertical, if the axis of x is horizontal. The reason for assuming this form for the equation of the arc $a\ b\ c$ is because it is the simplest form which contains *three* constants which can be determined so as to make it pass through the points a, b, c. We thus obtain for the area between the arc, the end ordinates and the axis of x, the formula—

$$\frac{h}{3}(y_1 + 4y_2 + y_3)$$

referred to by Mr. Macfarlane Gray as the "1, 4, 1" rule.

It will also be necessary in what follows to know the areas of the two halves A b and B c into which the parabolic element is divided by its middle ordinate. These, which we shall call the *half-parabolic elements,* are given respectively by—

$$h\frac{5y_1 + 8y_2 - y_3}{12} \text{ and } h\frac{5y_3 + 8y_2 - y_1}{12} \text{ respectively.}$$

Cubic Element.—If the base is *trisected* and y_0, y_1, y_2, y_3 are the successive ordinates, as good an estimate as we can form of the curve from these ordinates, and at the same time the most convenient one, will be obtained by supposing the equation of the curve to be of the form—

$$y = A + Bx + Cx^2 + Dx^3,$$

which represents a cubic curve, or the "parabola of higher order" of Mr. Macfarlane Gray's paper. Although Mr. Macfarlane Gray's point appears to be that this form was not originally considered by Simpson, it is shown in Todhunter's Integral Calculus that its adoption actually *does* lead to the so-called "1, 3, 3, 1" set of multiplies, the area being :—

$$\frac{3h}{8}(y_1 + 3y_2 + 3y_3 + y_4).$$

We shall, however, show that this formula is correct to the same approximation as the 1, 4, 1 formula, and not to a different order.

Application to the General Problem.—Now, every method of approximate quadrature consists essentially in dividing the area to be measured into n parts by equidistant ordinates, and measuring the lengths of the $n + 1$ ordinates (including the end ones), denoted by y_0, y_1, $y_2 \ldots y_n$. These ordinates are multiplied by certain factors which we shall call the *weights* of the ordinates, and the sum of the *weighted products* multiplied by h, the distance between each pair of ordinates, gives the area.

We notice, in the first place, that the sum of the $n + 1$ weights is equal to n (proved by putting all the ordinates equal when the area becomes a rectangle).

Calling w_0, w_1, $w_2 \ldots w_n$ the weights of the ordinates, the area is connected with these by the formula :—

$$\text{Area} = h(w_0 y_0 + w_1 y_1 + \ldots + w_n y_n).$$

In the trapezoidal rule, the elements of the area are taken as trapezoids, giving for the weights the values—

$$\tfrac{1}{2}, \quad 1, \quad 1, \quad 1, \quad \ldots \quad 1, \quad \tfrac{1}{2}.$$

The objection to this rule is that it makes the area too small if the curve is concave to the axis of x, too large if it is concave to the axis of y.

The form commonly given for Simpson's rule is based on the assumption that $a\,b\,c$ is one parabola, $c\,d\,e$ is another parabola, $e\,f\,g$ is a third parabola, and so on. This assumption requires that the number of elements shall be even and the number of ordinates therefore odd, and the weights of the ordinates are the series—

$$\tfrac{1}{3}, \quad \tfrac{4}{3}, \quad \tfrac{2}{3}, \quad \tfrac{4}{3}, \quad \tfrac{2}{3}, \quad \ldots \quad \tfrac{2}{3}, \quad \tfrac{4}{3}, \quad \tfrac{1}{3}.$$

Now the objection to this rule, which suggests itself, is that there appears to be no *prima facie* reason why one set of alternate ordinates y_1, y_3, y_5 . . . should have double the weight of the other set y_2, y_4, y_6

This objection receives support from the consideration that, if instead of $a\,b\,c$, $c\,d\,e$, . . . we had taken $b\,c\,d$, $d\,e\,f$ as the parabolic arcs proper, provision being made for the two ends, the even ordinates y_2, y_4, y_6, instead of the odd ones, would be of double the weight of the others. Moreover, as there is no particular reason why we should take one set of arcs as parabolas rather than the other set, we are led to the notion of adopting first one series of parabolas and then the other, and taking the mean of the two expressions for the area.

We are confirmed in this view by considering the particular case of a sinusoidal curve, determined by its maximum and minimum ordinates. Here, one set of parabolas will evidently make the area too large, the other will make it too small.

(*a*) *By using the trapezoidal formulæ for the end elements*, we get the following series of numbers for the weights—

$$\tfrac{1}{2}, \quad \tfrac{5}{6}, \quad \tfrac{4}{3}, \quad \tfrac{2}{3}, \quad \tfrac{4}{3} \quad \ldots,$$

and by taking the mean between these and the ordinary Simpson weights above found, we get for the weights—

$$\tfrac{5}{12}, \quad \tfrac{13}{12}, \quad 1, \quad 1 \quad 1, \quad \ldots \quad 1, \quad \tfrac{13}{12}, \quad \tfrac{5}{12},$$

a result which is true whether the number of ordinates is odd or even; for, in the case of an even number of ordinates, we only have to take the mean of two sets of weights, each of which corresponds to taking one end parabolic and the other trapezoidal.

(*b*) *By using the formulæ for the half-parabolic element* for the ends, we get the new series of weights—

$$\tfrac{5}{12}, \quad 1, \quad \tfrac{5}{4}, \quad \tfrac{2}{3}, \quad \tfrac{4}{3}, \quad \tfrac{2}{3}, \quad \tfrac{4}{3},$$

which, combined with the ordinary Simpson series, gives—

$$\tfrac{3}{8}, \quad \tfrac{7}{6}, \quad \tfrac{23}{24}, \quad 1, \quad 1, \quad 1, \quad \ldots \quad 1, \quad \tfrac{23}{24}, \quad \tfrac{7}{6}, \quad \tfrac{3}{8}.$$

(*c*) *By using the formulæ for the cubic element* for the portion included between the first four ordinates, and regarding the subsequent arcs through three consecutive points as parabolic (as explained above), we find the series of weights—

$$\tfrac{3}{8}, \quad \tfrac{9}{8}, \quad \tfrac{9}{8}, \quad \tfrac{17}{24}, \quad \tfrac{4}{3}, \quad \tfrac{2}{3}, \quad \tfrac{4}{3},$$

which, in combination with the ordinary Simpson series, give—

$$\tfrac{17}{48},\quad \tfrac{59}{48},\quad \tfrac{43}{48},\quad \tfrac{49}{48},\quad 1,\quad 1,\quad 1\ \ldots$$

(*d*) *By drawing intermediate ordinates bisecting the widths of the two end elements.*—We may call these intermediate ordinates $y_{\frac{1}{2}}$ and $y_{-n\frac{1}{2}}$. Applying the 1, 4, 1 rule to calculate the areas of the elements in question, and using Simpson's ordinary rule for the remaining portions of the curve, we get the following series of weights, those of the intermediate ordinates $y_{\frac{1}{2}}$ and $y_{n-\frac{1}{2}}$ being enclosed in square brackets—

$$\tfrac{1}{6},\quad [\tfrac{2}{3}],\quad \tfrac{1}{2},\quad \tfrac{4}{3},\quad \tfrac{2}{3},\quad \tfrac{4}{3},\ \ldots$$

or, by a combination of this method with the ordinary Simpson series:—

$$\tfrac{1}{4},\quad [\tfrac{1}{3}],\quad \tfrac{11}{12},\quad 1,\quad 1,\ \ldots\ 1,\quad \tfrac{11}{12},\quad [\tfrac{1}{3}],\quad \tfrac{1}{4}.$$

In order to test the relative accuracy of the various methods, I have applied them to the approximate calculation of $\log_e 5$ and $\log_e 10$ from the formulæ—

$$\text{Log}_e\, 5 = \int_{10}^{2} \frac{dx}{x};\quad \log_e 10 = \int_{10}^{1} \frac{dx}{x}$$

by means of the values of the ordinates corresponding to $x = 1, 2, 3, 4, 5, 6, 7, 8, 9, 10$. The following table shows the results, the correct values (taken from Todhunter's Algebra) being $\log_e 5 = 1{\cdot}60943791$, $\log_e 10 = 2{\cdot}30258509$.

Method.	Weights of Ordinates. y_0 / y_n	$y_{\frac{1}{2}}$ / $y_{n-\frac{1}{2}}$	y_1 / y_{n-1}	y_2 / y_{n-2}	y_3 / y_{n-3}	y_4 / y_{n-4}	y_5 / y_{n-5} &c.	$\log_e 5$ Correct Value. 1·609438	Error.	$\log_e 10$ Correct Value. 2·30258	Error.	Order of Approximation.
Trapezoidal Rule	$\frac{1}{2}$	—	1	1	1	1	1	1·62897	·01953	2·3790	·0764	h
Simpson's Rule	$\frac{1}{3}$	—	$\frac{4}{3}$	$\frac{2}{3}$	$\frac{4}{3}$	$\frac{2}{3}$	$\frac{4}{3}$	1·61084	·00141	Method cannot be used with an even number of ordinates		h^3
(*a*)	$\frac{5}{12}$	—	$\frac{13}{12}$	1	1	1	1	1·61600	·00657	2·3383	·0357	h^2
(*b*)	$\frac{3}{8}$	—	$\frac{7}{6}$	$\frac{23}{24}$	1	1	1	1·61242	·00299	2·3243	·0217	h^3
(*c*)	$\frac{17}{48}$	—	$\frac{59}{48}$	$\frac{43}{48}$	$\frac{49}{48}$	1	1	1·61141	·00197	2·3191	·0165	h^3
(*d*)	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{11}{12}$	1	1	1	1	1·61036	·00092	2·3104	·0078	h^3

It will be seen that Simpson's rule gives a result intermediate between those obtained by methods (*c*) and (*d*), but the example is rather a severe test of any rule, because the correct area between the first two ordinates is really $\log_e \frac{3}{2}$, and there is not much to choose between Simpson's rule and method (*c*). As a matter of fact, when we consider the order of approximation of the result, both are equally good, as I proceed to show in the next part of the paper.

Degree of Approximation of the 1, 4, 1, and 1, 3, 3, 1, Rules.

A good deal of misunderstanding exists as to the order of approximation to which the areas of curves are given by Simpson's and other formulæ.

If n be the number of parts into which the horizontal axis is divided, so that the total number of ordinates is $n + 1$, and a the whole distance between the bounding ordinates, the distance h between successive ordinates will be given by $n\,h = a$, and the order of approximation will depend on the lowest power of h which it is necessary to neglect in taking the approximate formulæ as representing the area.

Trapezoidal Rule.—Considering the strip bounded by two successive ordinates, take as origin of abscissæ a point midway between them, and suppose the equation of the curve between the limits of these ordinates to be given by the approximate formula—

$$y = \mathrm{A} + \mathrm{B}x + \mathrm{C}x^2 + \mathrm{D}x^3 + \mathrm{E}x^4 \;\ldots$$

The correct area between the bounding ordinates $x = \frac{1}{2}h$ and $x = -\frac{1}{2}h$ will be found to be—

$$\mathrm{S} = \mathrm{A}\,h + \frac{\mathrm{C}\,h^3}{12} + \frac{\mathrm{E}\,h^5}{80}, \text{ \&c.}$$

that given by the trapezoidal rule being—

$$\mathrm{S}_1 = \mathrm{A}\,h + \frac{\mathrm{C}\,h^3}{4} + \frac{\mathrm{E}\,h^5}{16}, \text{\&c.},$$

so that, going no further than h^3, there is a difference in the terms of the third order in h, represented by $\frac{1}{6}\,\mathrm{C}\,h^3$, and when all the elements are treated in this way, the percentage error in the area of the whole curve will be of the order of the ratio of h^2 to a^2, or, as we shall say, of the order of h^2.

The 1, 4, 1 *rule.*—In this case, taking the ordinates to be $x = -h$, $x = 0$, $x = h$, and the approximate equation of the curve within these limits to be the same as before, the area is found to be—

$$\mathrm{S} = 2\,\mathrm{A}\,h + \frac{2\,\mathrm{C}\,h^3}{3} + \frac{2\,\mathrm{E}\,h^5}{5}, \text{\&c.},$$

while that given by the 1, 4, 1 rule is—

$$\mathrm{S}_2 = 2\,\mathrm{A}\,h + \frac{2\,\mathrm{C}\,h^3}{3} + \frac{2\,\mathrm{E}\,h^5}{3}, \text{\&c.},$$

the approximate error being $\frac{4}{15}\,\mathrm{E}\,h^5$, and giving for the whole curve an error of the order h^4.

The 1, 3, 3, 1 *rule.*—Here we take the equations of the ordinates to be $x = -\frac{3}{2}h$, $x = -\frac{1}{2}h$, $x = \frac{1}{2}h$, and $x = \frac{3}{2}h$, and the area of the curve is—

$$\mathrm{S} = 3\,\mathrm{A}\,h + \frac{9\,\mathrm{C}\,h^3}{4} + \frac{243\,\mathrm{E}\,h^5}{80}, \text{\&c.},$$

that given by the 1, 3, 3, 1 rule being—

$$\mathrm{S}_3 = 3\,\mathrm{A}\,h + \frac{9\,\mathrm{C}\,h^3}{4} + \frac{63\,\mathrm{E}\,h^5}{16}$$

The approximate error is here $\frac{9}{10}\,\mathrm{E}\,h^5$, and is still of the order h^5, or for the whole curve, of the fourth order.

Applying these results to the methods here suggested, it is found that—

A NOTE ON SIMPSON'S RULES.

The ordinary trapezoidal rule is correct to the first power of h, but not to h^2.

Method (a) is correct to h^2, but not to h^3, as errors of this order are introduced into the two end elements, when these are treated as trapezoids.

Method (b) is correct to h^3, but not to h^4. The use of half parabolas at the two ends introduces errors of order h^4, which are of the same order as those produced by the 1, 4, 1 rule in the rest of the curves.

Methods (c), (d), (e), and the ordinary Simpson's rule are all correct up to h^3, but not to h^4, because they are all based on the 1, 4, 1 rule, or the 1, 3, 3, 1 rule. One method may give a rather smaller error than another, but there is no difference in the order of approximation of the methods.

It should be observed that the results given by methods (a) and (b), only differ by half the sum of the areas between the chords of the two first and last elements and their arcs, regarded as parabolic, and if it be decided to neglect this difference, the $\frac{5}{12}$, $\frac{13}{12}$ rule becomes applicable.

Method (a), moreover, if less accurate than the others, is very simple to use, as it merely requires the sum of ordinates used for the trapezoidal rule to be corrected by the addition of—

$$\tfrac{1}{12}\,(y_1 - y_0 + y_{n-1} - y_n)$$

As regards method (b), this can be employed by applying to the trapezoidal rule corrections represented by—

$$\tfrac{1}{8}\left\{ (y_2 + y_{n-2}) - (y_0 + y_n) \right\} - \tfrac{1}{6}\left\{ (y_2 + y_{n-2}) - (y_1 + y_{n-1}) \right\}$$

For method (c), the corrections to be applied become—

$$\left(\tfrac{1}{8} + \tfrac{1}{48}\right)\left\{ (y_1 + y_{n-1}) - (y_0 + y_n) \right\} - \tfrac{1}{12}\left\{ (y_2 + y_{n-2}) - (y_1 + y_{n-1}) \right\}$$
$$+ \tfrac{1}{48}\left\{ (y_3 + y_{n-3}) - (y_2 + y_{n-2}) \right\}.$$

For method (d), the corrections are—

$$-\tfrac{1}{4}(y_0 + y_n) + \tfrac{1}{3}(y_{\frac{1}{2}} + y_{n-\frac{1}{2}}) - \tfrac{1}{12}(y_1 + y_{n-1}).$$

I regret that this paper has extended to far greater length than I had originally wished, and, even in the present remarks, it has not been possible to discuss the application of Simpson's and allied formulæ to the areas of oval curves, where further difficulties present themselves in connection with the first and last elements. But I hope to have shown (1) that the 1, 4, 1, and 1, 3, 3, 1 formulæ represent the same degree of approximation, and (2) that, in measuring areas where a large number of ordinates are drawn, there is no degree of approximation obtainable by assigning unequal values to the odd and even ordinates in the middle of the curve which cannot be equally well obtained by applying to the trapezoidal rule a correction determined by three or four of the ordinates at the beginning and end of the curve whose area is required.

CONCLUDING PROCEEDINGS.

Sir NATHANIEL BARNABY, K.C.B. (Vice-President): Might I, my Lord, now ask that a hearty vote of thanks be given to the Society of Arts for the loan of their beautiful Hall? For more than forty years they have given us this hospitality. Among the many good works which they have done for the country, I think they may well reckon that of having given shelter to this Institution. We, ourselves, have done good work, but we have been helped greatly by the hospitality which the Council of the Society of Arts have accorded to us, and I propose that we give them our very hearty thanks.

Mr. BARNES: My Lord, I cordially second that proposal.

The resolution was put and carried unanimously.

Mr. H. M. ROUNTHWAITE (Member): My Lord, I am asked to propose a vote of thanks to the Council. As you know, they are all men working at high pressure; many of them come long distances to give us the benefit of their advice and experience, and I think we ought to be very much indebted to them, as indeed I know we all are. I have just one little remark to make—just a suggestion to throw out which may, perhaps, be considered hereafter. These are days of "spade work" and "efficiency," and our ordinary members are often put in an invidious position by being asked to decide between a retiring member of the Council and a new candidate, and they have no means whatever of knowing whether they are dismissing an old servant who has done excellent work, or whether they are dismissing a member who, from personal inertia or from pressure of business engagements, has not given sufficient attention to the requirements of his position. I suggest that if we had the number of attendances which each member of Council has given, shown in brackets after his name on the voting forms, it would assist us in our selection. I have only that suggestion to make, and I have very much pleasure in proposing a vote of thanks to the Council.

Mr. F. EDWARDS (Member): I have much pleasure in seconding that.

The resolution was put to the meeting and carried unanimously.

Mr. JOHN SCOTT, C.B., F.R.S.E. (Member of Council): My Lord, will you allow me to ask the members of the Institution here present to accord to your Lordship a very hearty vote of thanks for your conduct in the chair during this series of meetings. We are all aware that you hastened home from what was no doubt an extremely pleasant tour in Egypt, so that you might be able to perform your duties on this occasion. We think, therefore, that more than an ordinarily hearty vote of thanks is due to your Lordship for the earnest manner in which you have performed your duties on this occasion as our President. We have had many Presidents in this Institution, although not a very large number, considering the time of its existence. Sir Nathaniel Barnaby and some of my colleagues here have seen them all in office. I also am one of that number, but I do not know that we have ever had one filling the chair who has performed the duties of the office in a more genial and fitting manner, and therefore, my Lord, we desire to offer you our very hearty thanks.

Mr. S. W. Barnaby (Member of Council): Gentlemen, this motion, which has been proposed in such suitable terms by Colonel Scott, needs no words of mine to commend itself to you, but I hope his Lordship will forgive me if I repeat what was said to me this afternoon by one of our foreign members, because I think it accords with the feelings of us all. He said, "Your new President is splendid!" As this motion cannot be put from the chair, I ask you, gentlemen, to accord the vote of thanks in the usual way.

The motion was carried with acclamation.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, I have to thank you very cordially for the kind manner in which you have received the motion proposed by my friend Colonel Scott, and seconded by Mr. Barnaby. There is only one way in which I can place myself at all upon the same footing as the scientific men I see before me, and that is, that when I have a piece of work to do, I try to do it thoroughly, and, having taken on myself the high honour of being your President, I shall always, as long as I hold that position, do my best to fill it properly.

The proceedings then terminated.

PROCEEDINGS IN DÜSSELDORF.

VISIT OF THE INSTITUTION OF NAVAL ARCHITECTS

TO THE

SUMMER MEETING

OF THE

SCHIFFBAUTECHNISCHE GESELLSCHAFT.

JUNE 2, 3, 4, AND 5, 1902.

INTRODUCTORY PROCEEDINGS.

THE Summer Meeting of the Schiffbautechnische Gesellschaft took place at Düsseldorf on June 2 to 5, 1902. The members of the Institution having been cordially invited, through the President, Herr Geheimrath Busley, to take part in this meeting, a number of them (amounting, with the ladies of the party, to over 150 persons) took advantage of the invitation. The Council of the Institution were represented by the President, accompanied by Lady Glasgow, Lord Brassey, Dr. Elgar, Mr. J. I. Thornycroft, Professor Biles, Mr. Cornish, Mr. D. J. Dunlop, and Mr. Milton.

An informal reception took place on the Sunday evening in the smaller hall of the Düsseldorf Municipal Concert Hall, when Professor Busley briefly welcomed the guests, and notified one or two slight changes in the original programme.

The meetings, dinner, and reception were all held in the Stadtische Tonhalle (Municipal Concert Hall), a handsome building in the centre of the town, which afforded accommodation in its principal hall for the members and guests who had been brought together to the number of nearly 700.

The first meeting was held on Monday morning, June 2, at 10 o'clock, when H.I.H. the Crown Prince of Prussia, accompanied by Vice-Admiral von Tirpitz (Secretary of State for the Navy), arrived in the reception hall of the Municipal Buildings, where Professor Busley introduced the chief representatives of each nationality to His Imperial Highness. The Crown Prince then proceeded to the King's Hall, where he met with a most enthusiastic reception.

HIS IMPERIAL HIGHNESS, on behalf of H.M. the German Emperor, expressed His Majesty's regret at being unable to be present in person, as patron of the Schiffbautechnische Gesellschaft. His Highness then welcomed the assembled guests and declared the Congress open.

Professor BUSLEY (President of the Schiffbautechnische Gesellschaft) proceeded to deliver his inaugural address. He pointed out that this was the first time that their Society had held a regular Summer Meeting, at which they were able to welcome the representatives of allied institutions from other countries. He traced the progress of these institutions as illustrating the growth of the importance of naval architecture and marine engineering as factors in national welfare. Referring to England, Herr Busley paid a high compliment to the Institution of Naval Architects as the model upon which the others had been founded, and he laid stress on the debt which German naval architects owed to England for the lead she had given to other nations.

"It is only natural," he said, "that we should maintain the closest relations with the English Institution of Naval Architects: firstly, because, having been founded in 1860, it is the oldest of such institutions; and, secondly, on account of the great number of our countrymen, now occupying high positions in their native country, who have had a part, at least, of their practical training in England.

Many of these, in returning to their homes, have carried away the happiest recollection of England, and of the friendships they have made there."

Other speeches of welcome followed from Herr von HOLLEUFER, the Governor of the Rhenish Provinces; Burgomaster MARX, Mayor of Düsseldorf; Herr LUEG, Director of the Exhibition, and Herr CARP, President of the Rhenish Shipowners' Association.

The Earl of GLASGOW then rose to return thanks on behalf of the members of the Institution of Naval Architects:—

Your Imperial Highness and Gentlemen,—This is not the first occasion on which the Institution of Naval Architects has been made welcome on German soil; already, in 1896, both at Hamburg and Berlin, the Institution, under the presidency of my predecessor, Lord Hopetoun, experienced the great wealth of German hospitality, and many of those present to-day will doubtless recollect the splendid reception they received on that occasion. But in those days, although it is only a few years ago, your Society, as an Institution, did not exist, and the present gathering, with its promise of a most interesting and entertaining series of meetings and visits is an eloquent testimony to the wisdom of the founders of the Schiffbautechnische Gesellschaft, and the energy and thoroughness with which they have carried out their task. Your Society has had a singularly rapid growth; its inception may be said to have originated with that Summer Meeting of our Institution in 1896, as plans were, soon after that, formed for establishing it on lines similar to our own, although it was not until three years later that the scheme took final shape; and now you number, I am told, already nearly 1,000 members! But we have also a further and a closer tie, and that is the number of your members, and of your countrymen generally, who are members of our own Institution. These exceed by far the numbers of any other country represented with us, and, although you now have a similar establishment of your own, I trust I may not be too sanguine if I express a hope that we may in the future, as in the past, continue to number many and still more of our German friends among our members.

The gracious presence of His Imperial Highness the Crown Prince, representing, as he does, his august father, lends additional value to this meeting, by proving His Majesty's continued interest in the welfare of Institutions of Naval Architects and Marine Engineers all the world over, and in the work—of national importance—which they carry on. Indeed, the promotion of the cordial relations which exist between the various institutions of this kind, among the leading maritime nations, is a very gratifying result of the "Open Door Policy" (if I may use the expression) which has been pursued with regard to the interchange of professional knowledge and experience. If we, as the parent Institution, can claim to have inaugurated this policy, it is no less gratifying to find that you have followed on the same liberal lines, so that the undoubted benefits which accrue from such an interchange of ideas, and from the comparison of results of investigation and experiment, may be reaped by all alike.

The Exhibition, which is now being held here, and which we are looking forward to visiting in detail during the next few days, has already been recognised in the industrial world as a very remarkable proof of the activity of your local iron and steel industries and allied manufactures. That this great Exhibition should represent the industrial development of only two of the German Provinces is a very striking indication of the general progress of the German Empire, and we are very glad of this opportunity of seeing for ourselves the many exhibits of interest, the living proofs of the skill and industry of your Rhenish ironmasters and engineers, who have contributed so greatly to the prosperity of your country.

Monsieur NORMAND, Vice-President of the Association Technique Maritime, returned thanks on behalf of the French guests.

Professor BUSLEY then announced that the following telegrams would be despatched, if approved by the meeting:—

TEXT OF TELEGRAMS.

To HIS MAJESTY THE EMPEROR, *Potsdam.*

The shipbuilders and shipowners of America, Belgium, England, France, Holland, Norway, Austria, Russia, Spain, and Germany, now assembled in Düsseldorf, beg respectfully to thank your Majesty for having appointed His Imperial and Royal Highness, the Crown Prince of Prussia, to inaugurate their Congress, and they venture to express the hope that your Majesty will graciously continue in the future as in the past to take the same interest in the laudable objects which they have in view.

(Signed) BUSLEY, President of the Schiffbautechnische Gesellschaft.
GLASGOW, President of the Institution of Naval Architects.

To His Royal Highness, the GRAND DUKE OF OLDENBURG, *Oldenburg.*

The shipbuilders and shipowners of America, Belgium, England, France, Holland, Norway, Austria, Russia, Spain, and Germany, now assembled in Düsseldorf, venture to send your Royal Highness their heartiest good wishes for a safe return home, at the same time expressing their deep regret that your Royal Highness should be unable this day to occupy the presidential chair.

(Signed) BUSLEY, President of the Schiffbautechnische Gesellschaft.
GLASGOW, President of the Institution of Naval Architects.

The despatch of these messages having been unanimously agreed to, the President then called upon Herr Schrödter to read his paper.

THE IRONMAKING AND SHIPBUILDING INDUSTRIES IN GERMANY.

Abstract of Paper read by Herr E. Schroedter, Secretary of the German Iron and Steel Institute.

(Monday, June 2, 1902.)

This paper gives a sketch of the general development of the Iron and Steel Manufacture and of Shipbuilding in Germany, compares the extent of these industries with their present position in foreign countries, and discusses also some of the conditions which have had an influence upon their growth.

It is pointed out that the distribution of the coal mining, iron ore mining and pig iron making in Germany is not so favourable as in some other countries, as there are, in many cases, considerable distances between the coal basins and the ore districts; whilst, in all cases—especially as regards the important works in Upper Silesia—there are great distances over which the iron products have to be transported, in order to reach the sea coast. In spite of these disadvantages, the iron and steel trades of Germany have expanded enormously during the last decade; and, in view of the inexhaustible supplies of coal and ore known to exist in the German Empire, the future development of German industries is full of promise.

Some very instructive diagrams are given showing the total production of pig iron, during the thirty years extending from 1870 to 1900, in Germany, Great Britain, United States, France, Austria-Hungary, and Russia, and also showing the steel production in the four first mentioned countries from 1880 to 1900. In regard to pig iron, during the thirty years considered, Germany has always taken third place, the first place having been occupied by the United Kingdom, and the second by the United States, until 1889, when these two positions were reversed. While Germany still takes the third place as regards pig iron, she is now very little behind Great Britain, although, in 1870, her production was only one-quarter that of the British Isles.

Turning to the steel production, however, the recent growth has been relatively much greater in Germany, so that she now takes the second place, the United States being first and Great Britain third. This industry, as regards structural steel, may be said to have started towards the end of the seventies. Since that date the figures are:—

Year.	Germany.	Great Britain.	United States.
1880	614,000 tons	1,300,000 tons	——
1890	1,588,000 „	3,580,000 „	4,278,000 tons
1900	6,541,000 „	4,724,000 „	10,218,000 „

Not only has German steel making largely progressed in regard to the total output, but the actual capacity for the production of large pieces has more than correspondingly advanced. In 1880 only one firm in Germany could make plates 9 ft. 2½ in. broad, and only two others could deliver up to 7 ft. 6½ in.

and 6 ft. $2\frac{3}{4}$ in. broad respectively. Now plates can be made 11 ft. $9\frac{1}{2}$ in. broad, 65 ft. $7\frac{1}{2}$ in. long, by $1\frac{1}{2}$ in. thick.*

In heavy steel forgings an earlier development was made. So long ago as 1852 Krupp commenced making crucible steel forgings for carriage axles. In 1853 he made steel crank shafts for the Rhine-Cologne Steamshipping Co. The first steel propeller shafts were made in 1855 for the Danube Steamshipping Co., and four straight steel shafts, weighing altogether $22\frac{1}{2}$ tons, were made for the French Ministry of Marine, also in 1855. In 1861 the Hamburg-America Co. ordered one crank shaft of 4·14 tons and one straight shaft of $3\frac{1}{2}$ tons. In 1863, English firms commenced to order both crank and straight shafting of steel. Now, thanks to the use of powerful forging presses, shafts can be made of much larger dimensions, one being shown in the Düsseldorf Exhibition 150 ft. 3 in. long, having a finished weight of over 51 tons. This shaft shows very clearly that the German iron industry is ahead of the requirements of the shipbuilding trade.

Steel castings are claimed to be a German invention, the first having been made by Jacob Meyer, the founder and first technical manager of the Cast Steel Works of the Bochumer Verein, which were erected in 1843. The Essen Cast Steel Works took up the manufacture in 1862. In 1872, the Bochumer Verein cast a four-bladed ship's propeller 16 ft. 3 in. diameter, nearly 9 tons in weight, for the Hamburg-America Company. In the early seventies a cast steel works, now belonging to Krupp, was established in Annen, and towards the end of the seventies propellers, cross-heads, pistons, and other marine engine castings were made there for the home industry. In 1881 the first propeller brackets were made for a German builder, and in 1882 similar articles were sent to England. In 1886 the same works delivered the first steel sternposts for the Vulcan Yard at Stettin, and for the same vessels, steel castings were made for cross-heads, crank shaft bearings and hollow connecting rods. In 1888 another steel foundry was erected in Essen, and then began a large manufacture of marine constructive details, such as stems and sternposts, engine frames and columns, pistons, cylinder and slide chest covers, &c. From the middle of the nineties onwards this trade expanded, mainly owing to the exceptional demands made on it by marine engine builders, whose requirements have been the means of spurring on the steel makers to attempt and to succeed in their great performances of to-day, the high character of which is proved by the numerous exhibits in the Düsseldorf Exhibition. These also show that, at the present time, the manufacture of steel castings of all kinds is not confined to a few works, but is spread over nearly the whole country.

Armour plate manufacture was begun in 1876 by the Dillinger Hüttenwerke, who made the 8 in. wrought iron armour for gunboats of the *Wespe* class. In 1877, a compound armour came to the front, and in 1880 and 1881 the Dillinger Company commenced its manufacture, and continued making it until 1892, the thickest plates made being 15 in. In 1891 Krupp started a rolling mill for armour plates, commencing by manufacturing compound plates, but in 1892 these were given up, and unhardened nickel steel used instead. In 1893 a new quality of oil-tempered nickel steel of medium hardness was introduced. In 1895 Krupp first produced the hardened nickel steel plates which are now made by almost every armour-making firm in the world.

With regard to shipbuilding, Great Britain takes the first place by a long lead. Germany and the United States come next, there being little difference between these countries, in some years one, and

* In Mr. Sachsenberg's paper reference is made to a boiler plate exhibited by the Krupp firm, the dimensions being 87 ft. 11 in. long, 11 ft. $11\frac{3}{4}$ in. broad, and $1\frac{1}{2}$ in. thick. Also to two plates made by the Gutehoffnungshütte of Oberhausen from $65\frac{1}{2}$ ft to 69 ft. long. Besides these, the Hoerder Company exhibited two plates, one 74 ft. by 10 ft. 5 in., the other 82 ft. long by 7 ft. 9 in. wide, both being $\frac{11}{16}$ in. thick, and the Gewerkschaft Grillo Funke & Co. exhibited one plate 82 ft. by 6 ft. 6 in. by $\frac{3}{8}$ in. thick, the thinness of these increasing the difficulty of rolling.

in other years the other leading, but both are considerably ahead of France, which comes fourth on the list. Although Germany is a long way below Great Britain in point of tonnage, the total for last year being below that produced on one river of the North-East Coast of England, her shipbuilding has made a very great advance from a technical point of view, and in the case of the performances of a few vessels, she holds an unbeaten record. One reason assigned for the lateness of the development of the shipbuilding industry in Germany is the influence of classification societies. The Bureau Veritas was founded in 1830; Lloyd's Register was established in 1834, by the fusion of two organisations founded respectively in 1760 and 1799; but the Germanischer Lloyd only came into existence in 1867. Rules for the building of iron vessels were issued by Lloyd's Register in 1854, and by the Bureau Veritas in 1858, but it was only in 1877 that the Germanischer Lloyd published their rules. However, in spite of the lateness of its origin, the Germanischer Lloyd is successfully holding its own against the older competitors, as is shown by the records of the last twelve years.

Other and probably more important reasons assigned for the enormous lead of British shipbuilding are that in Germany the iron works lie far from the coast, whereas in England they are comparatively near the shipyards; and that in Germany the proportion of the iron material made which ultimately becomes used in shipbuilding is very small compared with the similar proportion obtaining in the United Kingdom, rendering it unprofitable for the German iron rolling mills to make the special sections required for shipwork. Another point is that the German sections were arranged for metric measures, whereas in England the sections were made in English measures, in accordance with the requirements of the rules for shipbuilding. These influences were so great that even when German shipbuilding began to expand, the sections were for a long time nearly all obtained from Great Britain, and it is only owing to a comparatively recent combination of German ironmasters, who agreed upon standard sections and undertook joint delivery and made a suitable division of the work among themselves, that the present use of German sections for German shipbuilding has become practicable.

In dealing with structural steel, it is natural that the rules for testing the quality should receive notice. The paper points out that the various bodies controlling these points, such as the Registration Societies, the various Admiralties, &c., all fix different limits of tensile strength for the material they approve of. This leads to difficulties where two or more of these bodies have to be satisfied at the same time, in some cases the requirements being so different that there is little or no common ground between them. The author advocates low tensile strength for all purposes, preferring the limits of $23\frac{1}{2}$—28 tons, and states that the elastic limit of steel of all degrees of hardness is exactly the same, and that, accordingly, within the strain limits permissible and usual, soft and hard metal behave in exactly the same manner.

DISCUSSION.

Herr Rudloff (Director of Naval Construction) raised the question of the quality of shipbuilding material. The controversy between steelmakers and shipbuilders was hardly, he thought, as acute as the author had pictured it. Owing to concessions on both sides, the situation had materially improved in recent years. In the Imperial Navy, for instance, the requirements of 1896 had not been adhered to, and for over eighteen months the specifications and tests required of steelmakers had been reduced in severity. For all parts not severely stressed, milder qualities of steel, tested up to 22 to 26 tons per square inch, were allowed to be used. As tensile strength is of the utmost importance in naval construction, it would be wiser to quietly await developments rather than to jump at sudden conclusions. This view, the speaker declared, was entertained by the North German Lloyd as well as by the Imperial Navy.

Chief Constructor WIESINGER, of Danzic, thought that the German Admiralty and the North German Lloyd had done more to meet the wishes of the steelmakers than had the English Lloyd's Register Society.

Herr EICHHOFF thanked the naval authorities for having in recent years consented to revise their test requirements. So long as England occupied her present leading position in the shipbuilding world, it would be necessary for German shipbuilders, if they wished to keep abreast of their English confrères, to adapt themselves to these conditions. The speaker proposed that a committee should be appointed to inquire into the whole question of quality of material, and to issue a report after evidence from shipbuilders and steelmakers had been received. In this way, the speaker hoped, the existing disadvantages would be removed, and the requirements lessened little by little.

Herr LUEG, of Düsseldorf, remarked that this was a very suitable place in which to raise this question, as the neighbourhood would furnish many representatives willing and anxious to take part in the proposed conference, the question of strength of materials being one of the highest importance to the district.

Herr MIDDENDORF and Herr KINTZLI also spoke in favour of simplicity and unification of tests.

Monsieur DAYMARD (Engineer-in-Chief of the Bureau Veritas, Paris) considered that Herr Schroedter, in recommending the use of mild in preference to hard steel for shipbuilding purposes, had raised the question of the suitability of converter steel, and, in the speaker's opinion, this material required very careful selection of its constituent parts to avoid brittleness, which is a very common drawback to its use. As regards the use of steel of high tensile quality, so much weight could nowadays be saved in this way that shipbuilders and designers would not readily give up employing it for warships and high-speed passenger vessels.

Professor BUSLEY (President) rose to assure the meeting that he felt convinced of the willingness of his Society to take up the proposal of Herr Eichhoff to appoint a committee to inquire into the matter.

The following are extracts from written contributions that have been made to the discussion on the above paper :—

Mr. J. T. MILTON (Member of Council, I.N.A.) points out, with reference to the differences that exist between the requirements of the registration societies and those of the various Admiralties, that these requirements do not have to be met at one and the same time. As regards the author's objections to the present fixed limits of strength because the results obtained from different test pieces are not identical, the writer considers that slight differences must necessarily occur as the result of small personal errors of observation and the want of absolute homogeneity in any given material, but with careful testers and good testing machines, these differences should be small enough to be negligible in practice. Commenting on the author's advocacy of the use of mild steel, on the ground that the elastic limit is the same for steel of all degrees of hardness, the writer considers that, even if this were the case, it by no means follows that the elastic limit of the test pieces is the useful limit of strength of the material in the structure. The splendid examples of metallurgical skill on view at the Düsseldorf Exhibition seem to show that German steelmakers, so far from taking the views expressed in the paper, are advancing in the direction of using larger and stronger pieces and building more and more powerful structures.

Mr. W. T. COURTIER-DUTTON (Member, I.N.A.) considers that as the purer qualities of pig iron become more difficult to secure, British manufacturers will find it more economical to adopt the basic open-hearth process. As regards the alleged hostility of Lloyd's to the use of basic steel, the writer mentions that steel so manufactured in America has for some time past been in use in Lloyd's vessels, and the British Corporation Registry have no objection to the use of basic steel provided it fulfils their prescribed test requirements. With regard to the standardisation of sections, the writer points out that in England this question is being actively dealt with by a committee representing the different industries affected.

MATERIALS AND MACHINE TOOLS FOR SHIPBUILDING AT THE DÜSSELDORF EXHIBITION.

ABSTRACT OF PAPER READ BY HERR GOTTHARD SACHSENBERG.

(Monday, June 2, 1902.)

GERMAN shipbuilding was, until about ten years ago, obliged to make use of English materials, for the simple reason that the English iron and steel works, with their long experience in the production of shipbuilding material, and their favourable conditions of production, were able to supply the German yards with more suitable and cheaper material than could be obtained at home, and in shorter time.

A reaction in favour of German shipbuilding material first set in when the German Admiralty set the example of having their vessels built in Germany, of German material. The requirements in regard to shipbuilding material, which till then had been kept within very modest bounds, now increased from year to year, so that the iron and steel industry, no less than that of machinery, and especially of machine tools, began to play an active part in German shipbuilding.

The object of this paper is to give those taking part in the Meeting a short description of some of the principal exhibits in the Exhibition in our branch of industry. It seems desirable to classify the exhibits, not according to the articles themselves, but according to the exhibitors. No account is given of naval artillery, or matters closely connected with it, as it would be impossible to do justice to the importance of the exhibits of this class in the paper. Neither is the interesting exhibit of Messrs. Felten & Guilleaume referred to, because it forms the subject of a separate paper.

The principal exhibits of the firm of Mr. Fried. Krupp, of Essen, are first described. These include a nickel steel armour plate of 104½ tons in weight, the largest plate that has yet been rolled. It is 43 ft. 1¾ in. long, 11 ft. 2 in. broad, and 11¾ in. thick.

The block from which this plate was rolled was 14 ft. 3½ in. long, 12 ft. 5 in. broad, and 3 ft. 4 in. thick, and 128 tons in weight. The plate was transported to the Exhibition upon a special 16-axled truck of 138 tons carrying capacity. Not less interesting is a boiler plate of the hitherto unequalled dimensions of 87 ft. 11 in. in length, 11 ft. 11¾ in. in breadth, and 1½ in. in thickness, weighing 29 tons. There are also a nickel steel six-throw crank shaft, 72 ft. 2 in. in length, weighing 112 tons, made for the North German Lloyd steamer *Kaiser Wilhelm II.*, with thrust shaft, intermediate shafting, and tail shaft with propeller all coupled together; and a straight piece of hollow shafting, 147 ft. 7½ in. long, 1 ft. 9¼ in. in outer diameter, and 4¾ in. inner diameter, weighing about 49 tons. This very long shaft was made as an example of what these famous steel works can produce. It was forged from a crucible steel block weighing about 79 tons. An historical account of the Krupp works and of its most interesting productions in the past is given.

The next exhibits are those of the Hoerder Bergwerk- und Hüttenverein in Hoerde, Westphalia. These include ship plates of great length and breadth, among which is an end plate for a boiler 11 ft. 9¾ in. in diameter, in one piece, and some large finished forgings of Siemens-Martin steel. The Bochumer Verein für Bergbau und Gussstahlfabrikation (the Bochum Mining and Cast Steel Manufacturing Company), one of the oldest cast-steel works, and the next in importance and power of output to those of Krupp, exhibits some large finished steel forgings of parts of marine engines, among which are a complete line of shafting for the Italian battleship *Regina Margherita*, some forged steel pistons, anchors, &c. The Rheinische Metallwaren- und Maschinenfabrik (the Rhine Metal and Engine Works) exhibits seamless steel tubes of all kinds and sizes for water-tube boilers, and seamless steam pipes up to

11¾ in. in diameter; and the Press- und Walzwerks-Aktiengesellschaft Düsseldorf-Reisholz, which is allied to this company, exhibit seamless tubes of large diameter and seamless boiler-plate rings, produced by the new Erhardt system of rolling. The Actien-Verein Gutehoffnungshütte, Oberhausen, employing 13,000 hands, and owning iron mines in Nassau, Siegen, Bavaria, Lorraine, Luxemburg, &c., over an area of 730 square miles, exhibit several finished lines of shafting, stern frame with rudder frame, stern and propeller of basic Siemens steel, and many other items. The Blechwalzwerk Schulz Knaudt (Schulz Knaudt Plate-Rolling Works) exhibit a quantity of material for land and marine boilers, among which are to be noted corrugated iron pillars made of short corrugated tubes welded together. The circumferential and the longitudinal seams are both heated by water-gas flames and brought together by machinery for welding, without any hammering, and the tube is afterwards corrugated by rolling. The largest of the tubes exhibited is 36 ft. 11¼ in. long, with an outer diameter of 3 ft. 11¼ in. and a thickness of $\frac{7}{16}$ in. The Duisburger Eisen- und Stahlwerke (Duisburg Iron and Steel Works) exhibit some large front plates for marine boilers and corrugated furnaces, and one front plate with corrugated furnace welded on to it. The Krieger Steel Works, of Düsseldorf, exhibit a stem, sternpost and rudder, engine seatings and pistons, &c. The German-Austrian Mannesmann Tube Works, of Düsseldorf, exhibit specimens of boiler tubes, steam pipes, and pillars made by their process. The Witten Cast Steel Works exhibit specimens of various sections of rails and bars, ship and boiler plates, of nickel and Siemens-Martin steel, and steel castings and forgings for marine engines. There are also exhibits in the same class by the Harkort Ironworks and Bridge Building Company, of Duisburg, the Hochfeld Rolling Mills, of Duisburg, and several other important establishments. Leurs & Hempelmann's Screw and Rivet Works, of Ratingen, exhibit specimens of their iron and steel rivets, and Franz Méguin & Co., of Dillingen, exhibit their perforated plates, waffle plates, and jointed chains. The Remscheid file and tool makers give a combined exhibition of their products. The firm of A. Mannesmann, of Remscheid, include in this some specimens of parts of engines made of their compound steel, which has a core of soft steel and outer working surfaces that are hardened by a special process. Messrs. Wenner Bros., of Schwelm, exhibit specimens of rivets and tools of various kinds. The Düren Metal Works exhibit specimens of their work in Durana metal and manganese phosphor bronze. The Benrath Engine Works, of Benrath, near Düsseldorf, exhibit a working model of a crane recently made for the Howaldt Works, Kiel, with a total outreach of 138 ft., and a lifting power of 148 tons at an outreach of 65 ft. 7 in., and designs and models of other electrically driven travelling cranes of large outreach and lifting power, winches, &c.

The exhibits in the Great Machinery Hall of the Exhibition are next described, which comprise shipbuilding materials and machine tools. The machine tool works of Ernst Schiess, of Düsseldorf-Oberbilk, contribute a horizontal ground lathe, with adjustable standards for working up to 31 ft. 2 in. in diameter, and 8 ft. 2½ in. in height; a planing machine, with a working length of 32 ft. 9½ in., and breadth of 13 ft. 1½ in.; and a threefold horizontal and vertical boring and slotting machine, for working up to 47 ft. 7 in. in length, 13 ft. 1½ in. in breadth, and 8 ft. 2½ in. in height. The Duisburg Engine Company exhibit many different examples of their products, including models and drawings of electrically driven cranes. Among the latter is a derrick crane, of 150 tons lifting power, made for the firm of Blohm & Voss, of Hamburg; one of 147½ tons for Mr. Krupp's Germania Shipyard, a floating crane of 29½ tons lifting power, with an adjustable outreach of 19 ft. 8¼ in. to 57 ft. 5 in., for the Hamburg-America Line of Hamburg, and arrangements of cranes for building slips. In conclusion, attention is called to the great power of production of the Rhenish Westphalian iron, steel, and engineering industries, which have succeeded in a comparatively short time, by their readiness to make sacrifices and by unremitting exertions, in gaining a position of importance commanding respect, not only in Germany, but throughout the world.

THE RIVER RHINE AND THE DEVELOPMENT OF ITS SHIPPING.

ABSTRACT OF PAPER READ BY W. FREIHERR VON ROLF.

(Tuesday, June 3, 1902.)

THE paper opens with a geographical and historical account of the Rhine from the earliest times. The author divides the river into four sections:—

(1) The Swiss Rhine above Bâle, which is only navigable for light traffic.
(2) The Upper Rhine, from Bâle, where navigation may properly be said to commence, to Bingen.
(3) The Middle Rhine, from Bingen to Cologne, the most mountainous part of the river's course, including, as it does, the "Bingerloch," cut through the solid rock.
(4) The Lower Rhine, from Cologne to the North Sea, through flat open country.

The tremendous power of the stream may be gauged from the fact that the theoretical energy of the great volume of water discharged by the river, in its fall of about 220 ft. between Bingen and the Dutch frontier, is calculated at more than 2,000,000 H.P. The regulation of the stream by training the banks and dredging the channel was absolutely essential to the development of traffic, and the latter has expanded in direct proportion to the care bestowed on the training of the river.

Frederick the Great, at the close of the Seven Years' War, laid the foundation of this development by establishing a Waterworks Administration, under whose care the river was placed. Much was done in the next ten years, but in 1794, the cession of the left bank of the river to France interrupted this work, and it was not until some twenty years later, and then under the newly formed governments of Cologne, Coblenz, and Düsseldorf, that it was resumed.

In 1851, the Rhine Conservancy Board was created by Royal decree, and then began a systematic improvement of the channel, the aim being to ensure a gradually increasing depth of 6 ft. 6 in. to 10 ft. between Bingen and the Dutch frontier, for a width ranging from 300 ft. to 500 ft. between the same points. The old arms of the river were closed up, the banks were protected, a towing path formed, sandbanks removed, and other improvements effected.

Steam dredgers were introduced about 1857, and, where the bed was rocky, steam boring machines and diving shafts were employed. Private dredging has also contributed to the improvement of the channel, for good gravel could be obtained, and this was readily removed for use elsewhere. The reef of rock at Bingen had always been the chief hindrance to traffic. The first widening at this place was begun in 1830 and further improved ten years later, but it was not until 1860 that a second channel was cut, thus enabling upstream and downstream traffic to keep clear of each other. The outlay on the river training works has been considerable, upwards of a million sterling having been spent between 1880 and 1899.

Notwithstanding the serious obstacles that had to be overcome, the Rhine bore considerable traffic on its waters, even before the close of the eighteenth century. Goods were carried in wooden sailing vessels of various types, capable of taking some 30 to 40 tons. As time went on, the sizes increased, and early in last century we find the larger vessels of the Middle Rhine capable of taking 150 to 200 tons, while, on the Lower Rhine, where the vessels were of the heavy Dutch type, their capacity would range up to 500 tons. Sails were used whenever practicable, but on the Middle and Upper Rhine towing by horses was in vogue when winds were contrary or the stream too strong.

In those early days the transport of say from 6,000 to 7,000 cwt. from Rotterdam to Cologne would be arranged as follows:—If the wind were favourable, the vessel would sail from Rotterdam to Emmerich in from three to four days. From there, from ten to fourteen horses would be harnessed to the vessel, which would reach Cologne in from ten to eleven days. The downstream voyage from Cologne to Rotterdam was made in seven or eight days; sometimes even in six days. For the transport of about 2,000 cwt., the distance from Cologne to Mainz would be covered, with a full river, in fifty-two hours, or, with a low river, in from seventy-eight to eighty hours.

In the matter of dues, the shipping of the Rhine has never been spared. At the beginning of the Middle Ages, these were looked upon as payment for the removal of obstacles to the traffic. It was after the Rhine dues had ceased to be entirely in the hands of the German kings, and after the lords of the various territories, near the banks of the river, began to levy their own tolls, that what was at first a help to the traffic became more and more a hindrance to it.

There is an old saying that illustrates the situation:—

> **They claim their dues, king, bishop, priest,**
> **And lords of many ranks,**
> **Until the long Rhine counts more tolls**
> **Than miles along its banks.**

The burdens thus imposed upon shipping increased as time went on, and threatened its very existence. It was not until 1803, however, after France, by the Treaty of Lunéville, had acquired an interest in the river, that the existing dues were abolished and an equitable system established—a first step towards freeing the river navigation altogether. The passenger traffic on the Rhine, as early as the beginning of the eighteenth century, was already well established, it being in the hands of no fewer than fifty-one shipmasters, who controlled the traffic on the different portions of the river; and so great was the competition that frequent disputes arose.

With the utilisation of steam power for the propulsion of vessels, began an entirely new epoch for the Rhine. The first steamboats that plied upon the river were those of two Englishmen, Wager and Watt, the former trading with the *Prinz von Oranien*, and the latter with the *Caledonia*. It is known that the *Caledonia* was fitted with a pair of engines of 50 H.P. each, and that she was propelled by side paddle wheels. James Watt sailed in his vessel across the Channel in the year 1817 into the Scheldt and Maass, and up the Rhine to Coblenz. But the results were not satisfactory, and it was not until 1825 that a regular steamship service was established between Cologne and Rotterdam, from which port a connection was made with London by the steamer *Batavier*, fitted with low-pressure engines of 200 H.P. Various companies were formed to exploit the new means of propulsion, and it was not long before the old shipmasters, interested in sailing vessels only, were hard put to it to meet the competition from the steamers, which they declared to be quite unsuitable for river traffic.

In 1829, the Netherlands Steamship Company experimented with the S.S. *Herkules*, a steamer capable of carrying 100 tons of cargo and at the same time towing some half-dozen barges. Though the venture did not prove remunerative, it was the precursor of other and more successful enterprises. The Prussian-Rhine Company, which started about this time, proved the practical possibility of making steam pay on the river, and, at the end of the first ten years of its existence, it had transported a million passengers and 100,000 tons of goods. Other companies followed suit, and the efficiency of the steamers improved as the numbers increased. In 1838, a direct service between Rotterdam and Mainz was established, the return journey between Düsseldorf and Mainz being accomplished in 33½ hours.

About 1840, iron began to displace wood for the construction of the hulls of steamships, the first Rhine vessel entirely constructed of iron being the *Victoria*, built and engined in England. Forty years later, this vessel was reconstructed and fitted with high-pressure engines with oscillating cylinders (high and low pressure) and new paddle wheels with feathering floats.

With the improvement in the steamers and the increase in the number of companies interested, competition became keener, and the fares and the duration of the voyages were materially reduced. The two chief companies, with headquarters in Cologne and Düsseldorf respectively, finally amalgamated in 1853, and, with their combined resources, greatly improved the facilities both for passengers and goods on the Rhine. A new factor now entered into play, namely, the railway competition along the banks of the Rhine. This stimulated the steamship companies to fresh efforts, and in 1866, two saloon steamers, the *Humbolt* and the *Friede*, were constructed on the American plan for passengers only, and put into regular service. These were added to in succeeding years, and the company now possesses over thirty such vessels.

The development of the towing traffic proceeded side by side with that of the passenger traffic. The Mannheim Steam Tug Company, organised in 1863, started with a fleet of four paddle tugs and six barges, carrying annually 2,300 tons. They now possess twelve screw tugs and over sixty-one barges, and their annual traffic is nearly 60,000 tons. Other companies show a similar development, and the author traces the progress of the chief concerns engaged in the transport of goods on the Rhine.

In 1876, the Fixed-Rope Towing Company was established, the towing being done by means of a permanent cable laid along the bed of the river, wherever the current was considerable. For the downstream journey these vessels use their twin screws, dispensing with the fixed rope. This company handled over a million tons of goods in 1901.

Tables are given showing the aggregate tonnage handled on the Rhine between 1835 and 1900, and also showing the number of bridges and other hindrances to the freedom of traffic which exist in the river.

The paper is throughout illustrated with numerous photographs and drawings, descriptive of the various types of vessel engaged in the Rhine navigation from the earliest times to the present day.

DISCUSSION.

Time did not allow of any discussion at the conclusion of this paper, but a contribution has been received from Mr. G. C. Mackrow (Member I.N.A.), of which the following is an extract:—

The paper is of particular interest to one who, like the writer, has been long connected with the Thames Ironworks Company, as several of the earliest Rhine steamers (such as the *Victoria* and the *Mannheim No. I.*) were constructed by this firm. At the launching of the latter vessel, when the writer was present, the side plates between the frames buckled as the vessel took the water, but when wholly afloat, they again assumed their normal form ; a remarkable fact, as the vessels in those days were built of iron and not of mild steel, but the materials were very carefully selected. From the year 1839 to 1844, the writer's firm built altogether seven vessels for the Rhine navigation.

MARINE WIRE ROPES.

Abstract of Paper read by Herr F. Schleifenbaum.

(Tuesday, June 3, 1902).

For centuries hemp ropes and iron chains have been chiefly employed in shipping and other industries as a means of traction, and it is only comparatively recently, since the introduction of wire ropes, that the special properties of iron and steel wire have been utilised in this direction.

Wire ropes are superior in many respects. If the wires are galvanised as a protection against rust, the rope is better able to withstand the pernicious effects of atmospheric influences. A hemp rope retains the damp internally, no matter what care is taken to prevent it, and it will rot from the inside without giving any warning from its outward appearance. Another advantage of the wire rope is its greater tensile strength as compared with that of a hemp rope of equal diameter; so that, for an equal strain, the wire rope may be selected of smaller diameter, thus presenting less resistance to the wind in the rigging, besides economising space if kept on deck or stowed away. As compared with chains, wire rope is also superior in that it occupies less space, and is much stronger weight for weight. It is also less liable than a chain to danger from sudden breakage, for the weakness of any link whose failure renders the chain useless is very seldom detected beforehand, while a wire cable shows by the rupture of a single strand (whose broken ends protrude) that danger is at hand.

The author proceeds to trace the early history of the introduction of iron wire into the manufacture of cables. He ascribes to Albert of Clausthal the honour of having, in 1831, been the first to substitute wire ropes for chains at the mines in the Harz Mountains. In these early ropes, the wires and strands were both laid in the same direction. In the beginning of the forties Théodore Guilleaume introduced wire ropes with a cross-lay (the wires and strands laid in opposite directions), and. to render the ropes more flexible, he provided them with a hemp centre. Since that time, many new forms of construction have come to the front, some of them of considerable importance for particular purposes, but in its general application the cross-lay with hemp centre has proved the greatest improvement in wire ropes, as it renders possible the manifold uses to which wire ropes are applied at the present day. For marine purposes, the cross-lay has become the prevailing construction.

It was not until some twenty years later that wire ropes began to be extensively used for marine purposes, as so much distrust of their reliability had first to be overcome. In 1883, steel wire ropes were adopted exclusively for standing rigging in the German Navy, and from this time the extension of the use of wire ropes became very marked: for hauling, towing, and mooring, as well as for numberless purposes aboard ships, these ropes came to be generally employed.

The author discusses the various conditions under which wire ropes are used, the strains to which they are subjected, the chief causes of wear and tear, and the best means of meeting these conditions and of prolonging the life of cables. Galvanising the wires and frequent rubbing with linseed oil are recommended as a protection against rust, and attention is called to the dangers arising from kinks in handling, or from galvanic action by contact with bronze in the presence of sea water. Details of construction are described—the gauge of the wires employed and the kind of lay adopted, these being governed by the use to which the rope is to be put and the amount of flexibility required. The use of aluminium bronze wire is recommended where the risk of rusting is great, but its high cost and low tensile strength are objections to its general use.

Wire rope is also largely employed for towing purposes on navigable rivers. On portions of the Rhine where the current is strong, a heavy cable is laid along the bed of the river by means of which the tugs haul themselves and their strings of barges along. Through the narrow "Iron Gates" of the Danube a similar system is in vogue, the rope itself being a "locked coil," composed of wires so shaped that they interlock on their outer faces, so as to form an almost complete metallic cross section. For the traction ropes of ferries the well-known hawser-lay is usually employed, but "flattened strand" ropes have lately been adopted, and they are also used by shipbuilders for winding and crane ropes. As their name implies, the separate strands are of flat oval section, instead of being cylindrical like those of the old constructions. Other special forms of ropes are described, such as floating hawsers, partly composed of cork, and ropes for winding round steam pipes to strengthen them, and for protecting lead water pipes from external injury. The testing of wire ropes is provided for by the specifications of the German Lloyd, and these require that, when the wire is tested, its breaking strain multiplied by the total number of wires composing the rope must exceed the specified breaking strain of the rope by at least ten per cent.

DISCUSSION.

Chief Constructor HÜLLMANN, of Kiel, remarked that one great advantage of steel wire rope over chain cables lay in its behaviour when breaking. A chain, in snapping, flies in all directions, tearing through everything its meets: while the wire cable breaks so quietly that one can stand in the neighbourhood of a highly strained wire rope without danger.

Herr MIDDENDORF, of Berlin, a director of the Norddeutscher Lloyd, was of opinion that the time had not yet come to substitute wire rope for anchor chains.

Dr. F. ELGAR (Vice-President I.N.A.) asked at the close of the discussion to be allowed to say a few words on behalf of the British guests, in acknowledgment of the kind welcome that had been extended to them. As a proof of the trouble that had been taken on their behalf, he mentioned the translation into English of all the papers read at the meeting, and he felt sure that this thoughtful act of courtesy had been much appreciated. As regards the Exhibition, and the visits to the works that had been thrown open to inspection, the speaker said that the wonderful proof of the skill of German engineers and manufacturers was no surprise to those who had followed the remarkable development of the iron and steel industries in that country during recent years. The foreign guests would offer to their German colleagues their hearty congratulations on the success of the Exhibition, and their warmest thanks for the splendid reception that had been extended to them.

CONCLUSION.

THE following is the text of the letter, conveying the vote of thanks of the Council of the Institution to the President and Members of the Council of the Schiffbautechnische Gesellschaft:—

INSTITUTION OF NAVAL ARCHITECTS,
5, ADELPHI TERRACE, LONDON, W.C.,
July 16, 1902.

Geheimer Regierungsrath Professor BUSLEY,
President of the Schiffbautechnische Gesellschaft.

MY DEAR SIR,

At a recent meeting of the Council of this Institution it was unanimously resolved to pass a very hearty vote of thanks to yourself, and to the Members of the Council of the Schiffbautechnische Gesellschaft, for the cordial and friendly welcome which they extended to those Members of this Institution who had availed themselves of your invitation to visit Düsseldorf, and take part in your recent Summer Meeting.

Nothing could exceed the kindness and hospitality shown to our Members on this occasion, and all of those who took part in this trip experienced the liveliest satisfaction, both at the heartiness of their welcome and at the variety and interest of the visits and excursions that had been arranged for their benefit, and which were so admirably carried out.

The Council wish specially to record also, on behalf of the ladies of their party, their appreciation of the active part which the Ladies' Reception Committee took in organising and carrying out the various entertainments for the benefit of the lady guests, and which added so much to the general success of the visit.

Let me conclude, my dear Sir, by recording the pleasure it gives me to convey this resolution to you in the name of our Institution, and, with sincere expressions of esteem and regard.

I remain,
Yours very truly,
(Signed) GLASGOW,
President, I.N.A.

OBITUARY NOTICES.

SIR JAMES LAING, J.P., D.L.

The death of Sir James Laing, which occurred on December 15, 1901, at his residence, Etal Manor, Northumberland, at the age of seventy-eight, has removed one of the best-known figures in the shipbuilding world of the North of England, and, indeed, of the country.

Born in Sunderland in 1823, James Laing began his shipbuilding career in his father's yard, which had been established in the neighbourhood for some years previously. At the age of twenty he was made manager of the works, and from that time forward the success which attended the firm was largely attributable to his energy and business ability. In those early days, ship construction was solely confined to wooden vessels; but a few years after Mr. James Laing had taken over the management of the works, his attention was called to the remarkable evidence of the strength of Brunel's iron vessel—the *Great Britain*—which had been wrecked off the coast of Ireland, and had withstood a winter's gales in a very exposed position. Iron had evidently come to supersede wood, and Mr. Laing resolved that his firm should not be behind the times. Their first iron vessel was accordingly launched in 1853, and the output of the firm steadily increased, until, in 1900, over 40,000 tons of new tonnage was launched, a total far exceeding that of any other firm on the Wear.

Sir James Laing's energies, however, were not confined to shipbuilding alone. He took a very prominent part in all local affairs, serving for no less than thirty-two years as chairman of the River Wear Commission. He was an ardent politician, though as a candidate for Parliamentary honours he was not successful.

In 1883 Mr. Laing was elected President of the Chamber of Shipping of the United Kingdom, and, as such, he ably championed the cause of the shipowners in securing a revision of the regulations and charges of the Suez Canal, of which company he eventually became director.

The honour of knighthood was conferred on Sir James Laing in 1897.

His connection with the Institution dates from 1884, and when, three years later, the Summer Meeting of the Institution was held at Newcastle and Sunderland, Mr. Laing took a prominent part in welcoming the members to his native town.

In 1885 he was elected to a seat on the Council of the Institution, which he occupied until his death.

BENJAMIN MARTELL.

The death of Benjamin Martell removes from the roll of the Institution one of its most prominent and most popular members.

Mr. Martell was born in 1825, and died at his residence at Blackheath on July 15, 1902. He served his apprenticeship in the Royal Dockyard, Portsmouth, at the time when Mr. John Fincham was master-shipwright. He assisted his chief—who was himself an original member of the Institution, and one of the first Members of Council—in the preparation of his work on Naval Architecture. Leaving the public service, he joined Mr. C. Lamport, another original member of the Institution, and became manager of the latter's shipyard and repairing works at Workington. In 1856 he entered the service of Lloyd's Register of Shipping, becoming an assistant surveyor at Sunderland. Subsequently he was appointed as surveyor to the ports of Greenock, Southampton, Leith, Sunderland, and North Shields. In 1872 he succeeded Mr. Bernard Weymouth as Chief Surveyor, which position he held until his retirement three years ago.

Mr. Martell joined the Institution in 1873, was elected to the Council the following year, and became Vice-President in 1887. His contributions to the Transactions of the Institution have been exceedingly numerous, and only the more important ones can be here noted. The first paper was contributed in 1874, and was "On Freeboard," a subject with which his name has been prominently connected. Three years later he read a second paper, "On Water Ballast"; and in the following year, 1878, in a contribution, "On Steel for Shipbuilding," he brought before the Institution one of the great questions of the day. In 1886 he presented another paper on the same subject, "A Review of the Progress of Mild Steel, and the Results of Eight Years' Experience of its Use for Shipbuilding Purposes"; and again in 1888 he reviewed another aspect of the question by his paper "On the Present Position Occupied by Basic Steel as a Material for Shipbuilding." His paper of 1874, on Freeboard, was followed by another in 1880, "On Causes of Unseaworthiness in Merchant Steamers," which has done much to increase the safety of human life at sea; and two years later he read his notable paper "On the Basis for Fixing Suitable Load Lines for Mercantile Steamers and Sailing Vessels." Ten years later, in 1892, his paper "On the Alterations in the Types and Proportions of Mercantile Vessels, together with Recent Improvements in their Construction and Depth of Loading as Affecting their Safety at Sea," again dealt with a question upon which he spent so much time and labour. It would be difficult to over-estimate the far-reaching importance of these labours in their influence upon mercantile shipbuilding of to-day.

Mr. Martell was so constant an attendant at the meetings of the Institution, and so frequent a speaker during discussions, that his life's work may almost be gathered from our Transactions. His great work has been that connected with the question of freeboard. The year in which he joined the Institution, Parliament appointed the "Commission on Unseaworthy Ships," and his first paper, to which reference has been made, was the result of this step. In 1875 the Board of Trade invited the Committee of Lloyd's, and of the Liverpool Underwriters' Registry, to assist in stating principles upon which rules as to freeboard might be based. This was found to be a difficult task, but Mr. Martell's paper of 1882 ultimately had the effect of leading to the adoption of certain rules which were accepted by the Load Line Committee of 1885, of which Sir Edward Reed was President. It was recommended in the report that the Committee of Lloyd's Register should administer the load-line regulations, and this recommendation was carried into effect by Act of Parliament in 1890.

On Mr. Martell's retirement from the Chief Surveyorship of Lloyd's Register, a committee was formed to present him with a testimonial, and the greater portion of the sum which had been contributed Mr. Martell generously placed at the disposal of the Institution, for the purpose of endowing a scholarship in Naval Architecture. The offer was gratefully accepted, and regulations were drawn up for the administration of the Scholarship.

It is a sad coincidence that the first year of the award of this Scholarship should be that of the death of its founder; but, although he did not live to see the actual result of his liberality, his name will thus be associated with the Institution which owes so much to his unremitting labours, and his unselfish interest.

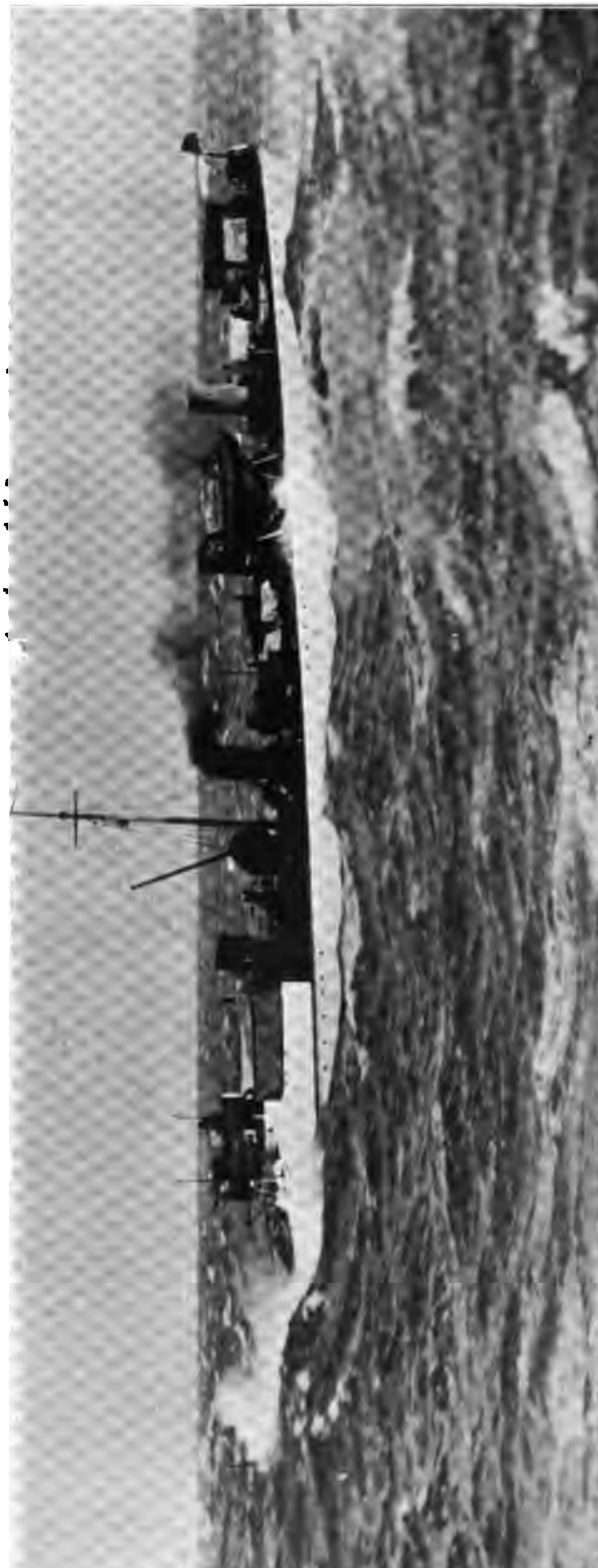
His memory will ever be held in affectionate regard by his colleagues on the Council, and by the members of the Institution.

NOMINAL INDEX.

BEING AN ALPHABETICAL LIST OF THE NAMES OF THE AUTHORS OF PAPERS READ BEFORE THE INSTITUTION, AND OF THE SPEAKERS WHO TOOK PART IN THE DISCUSSIONS WHICH FOLLOWED THE READING OF THE PAPERS.

UNWIN BROTHERS, LIMITED, LONDON AND WOKING.

To Illustrate Mr. S. W. Barnaby's Paper on "Torpedo Boat Destroyers."



[Photo. by Messrs. West & Son, Southsea.

FIG.

Curve of the tensions and de
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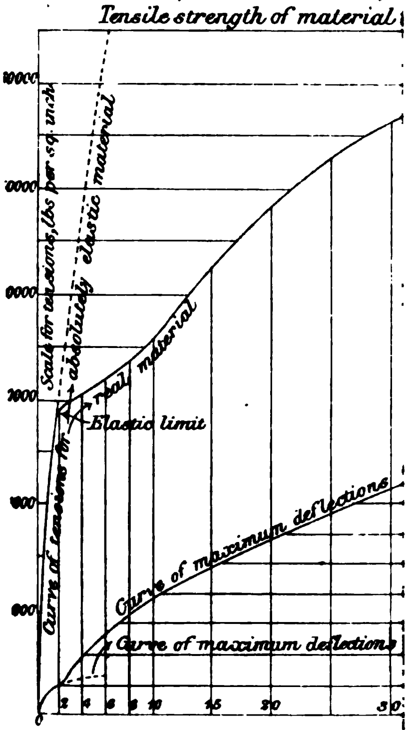
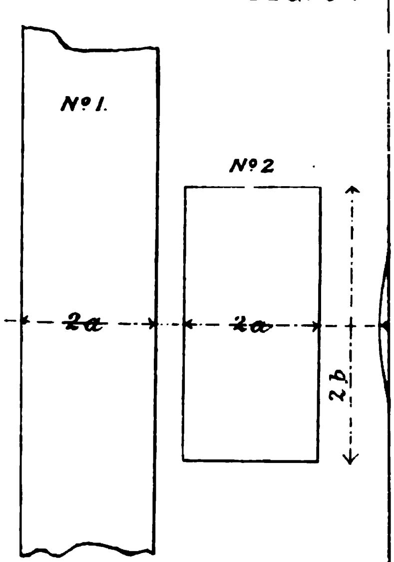
(The plate being supposed
Span 2a-100 inches, a
Tensile strength of material



Total load

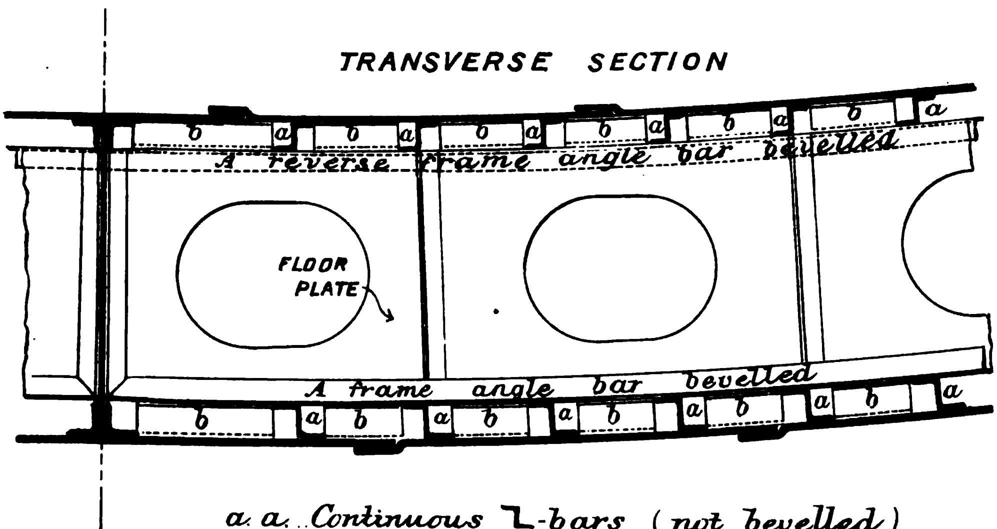
FIG. 3.



To Illustrate Mons. I. G. Boobnoff's Paper on "The Stresses in a Ship's Bottom Plating due to Water Pressure."

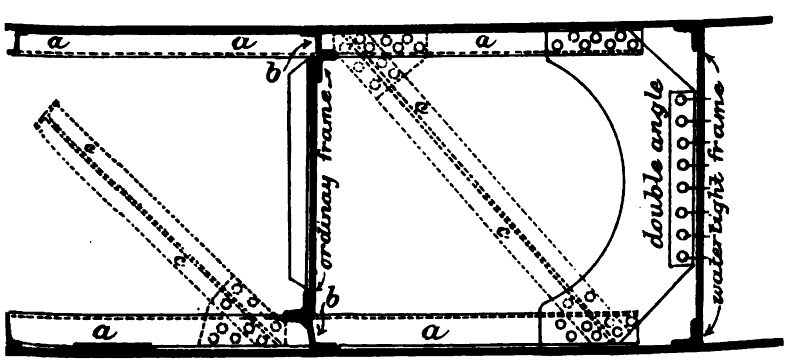
FIG. 6.

TRANSVERSE SECTION



a. a. Continuous L-bars (not bevelled)
b. b. Short L-strips (not bevelled)
c. c. Diagonal stays (tee section) over 6 - 12 feet.

LONGITUDINAL SECTION



PLAN

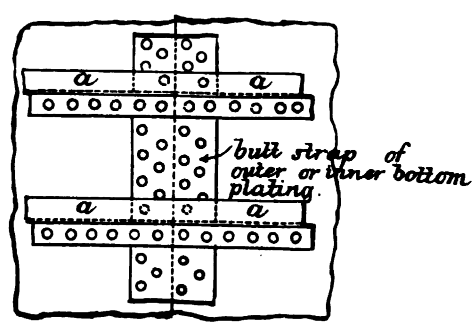


FIG. 7.

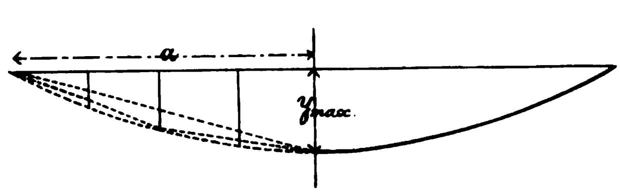
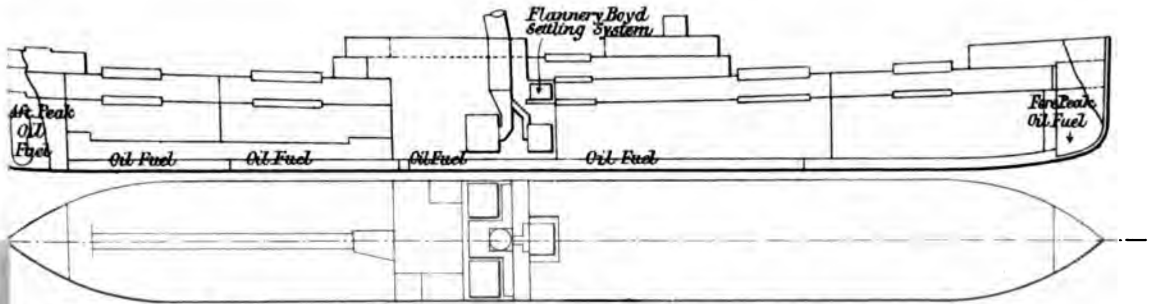
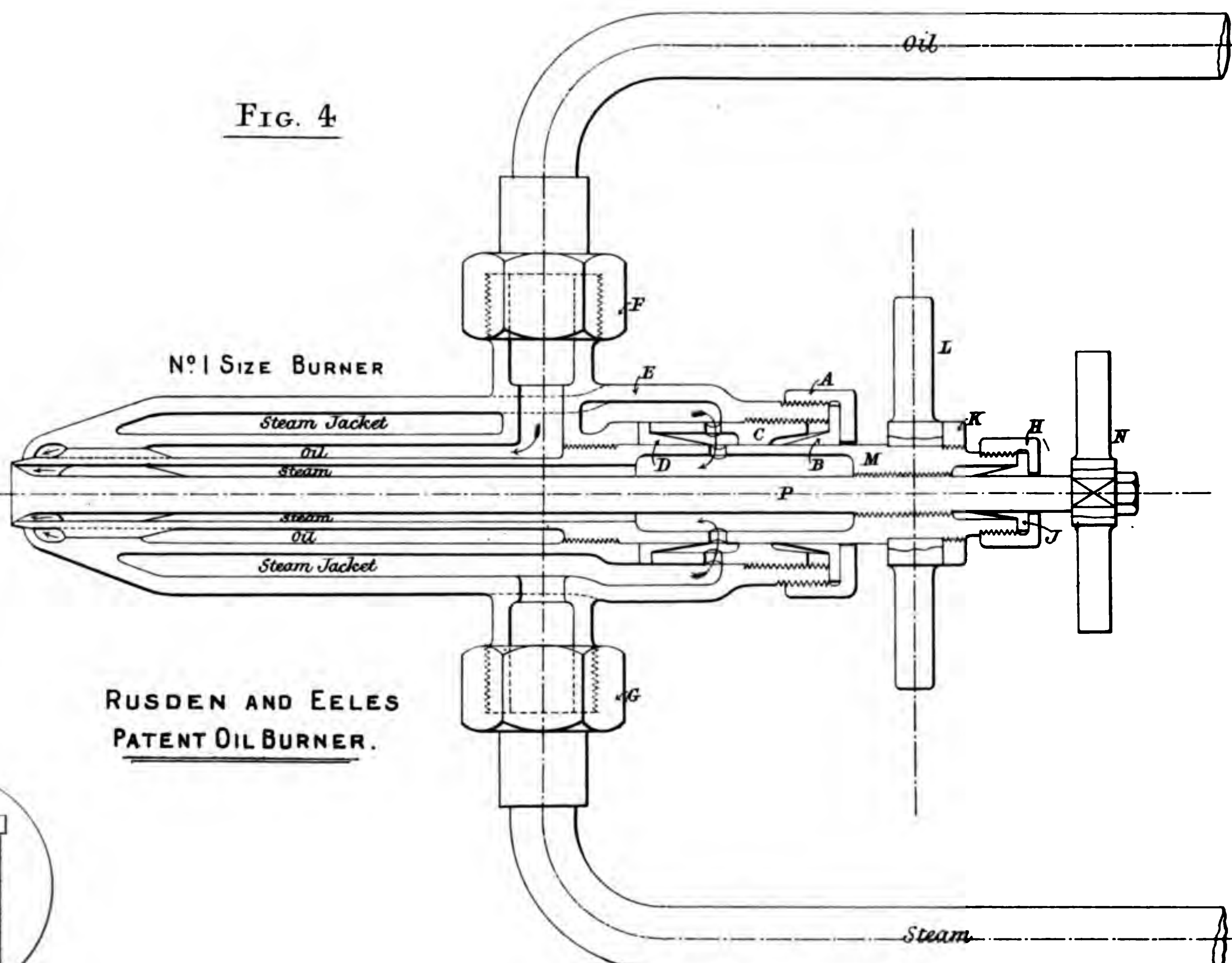


FIG. 2.

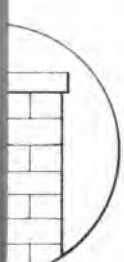


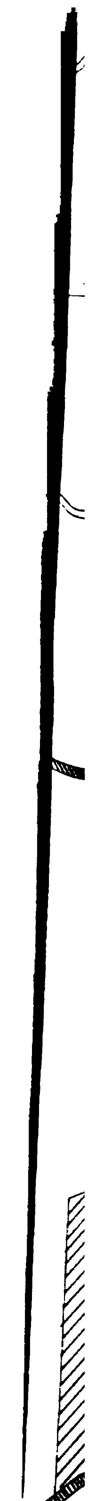
STORAGE OF OIL FUEL IN ORDINARY CARGO STEAMER ON THE FLANNERY-BOYD SYSTEM.

FIG. 4



RUSDEN AND EELES
PATENT OIL BURNER.





e bricks:

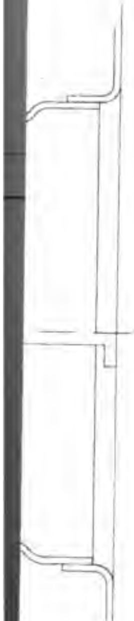


Fig. 9.

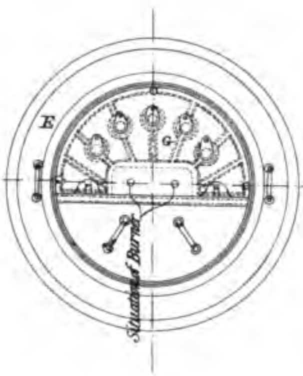
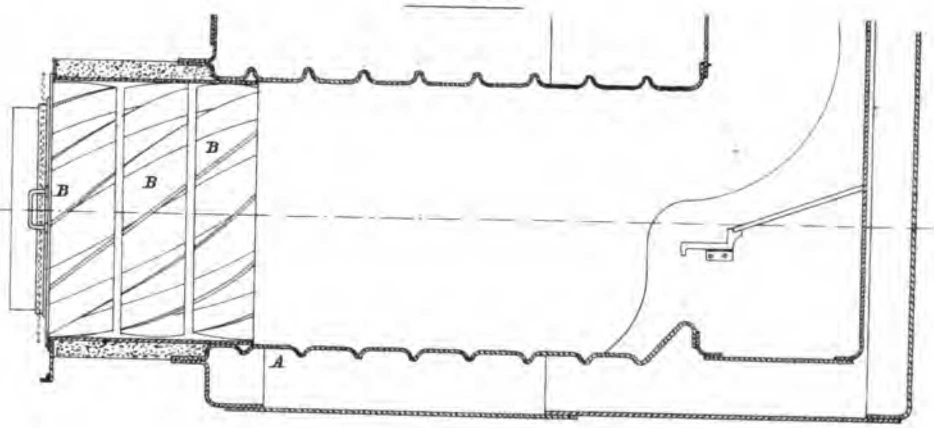


Fig. 10.



DETAILS OF FURNACE
ON
THE "MEYER" SYSTEM.

Fig. 11

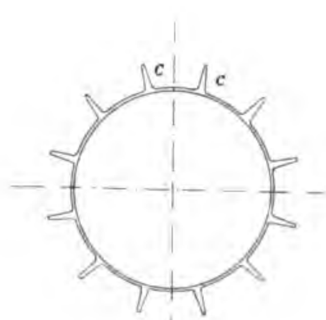
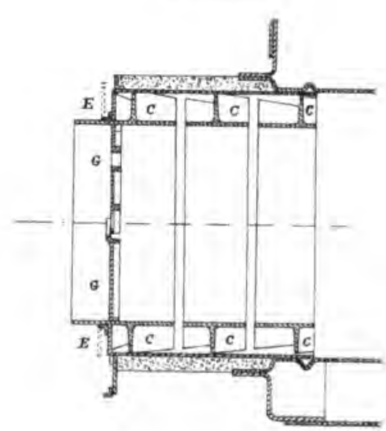
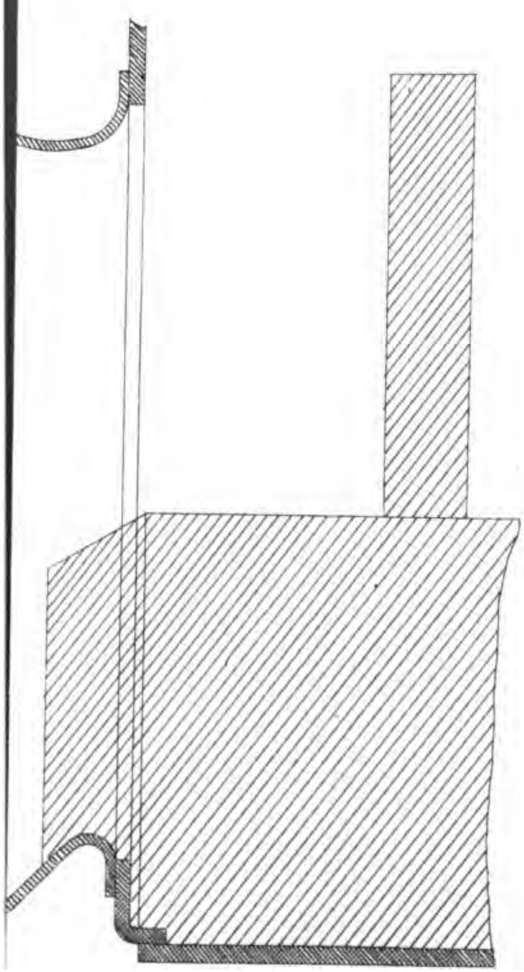


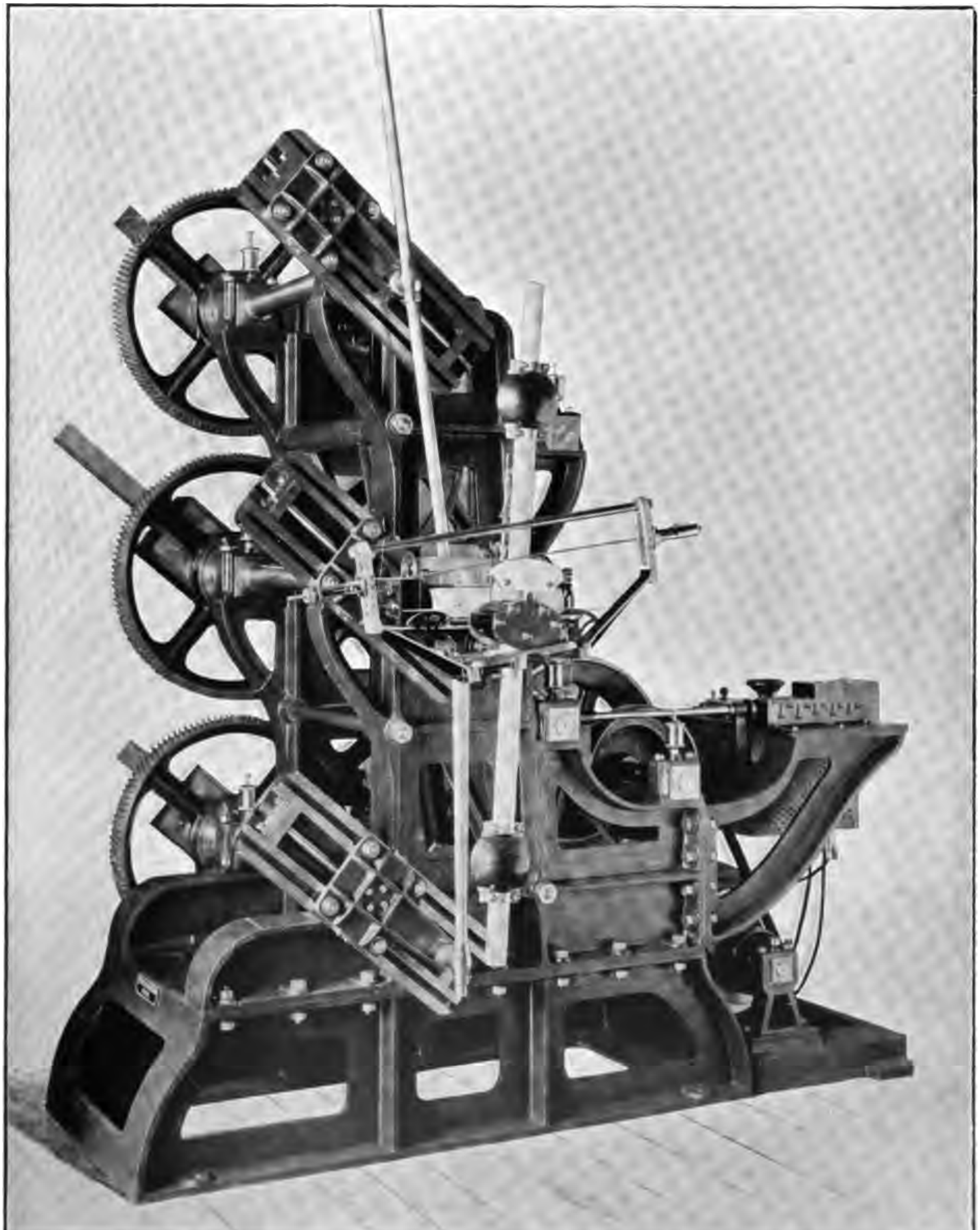
Fig. 12.

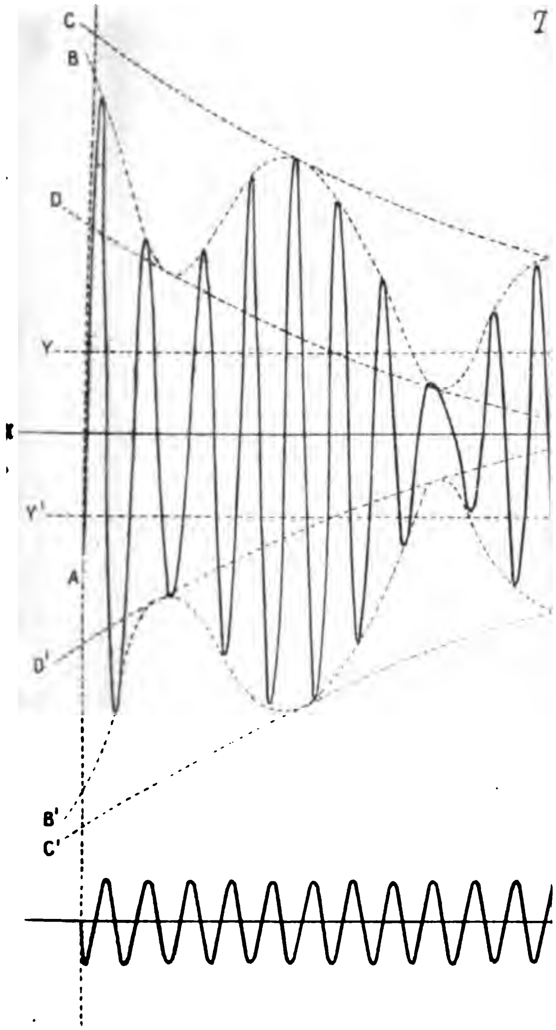




bricks in way of saddle plate.

To Illustrate Captain G. Russo's Paper on "The Navipendular Method of Experiments, as applied to some Warships of different classes."

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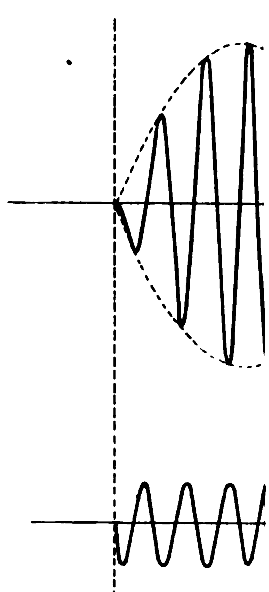


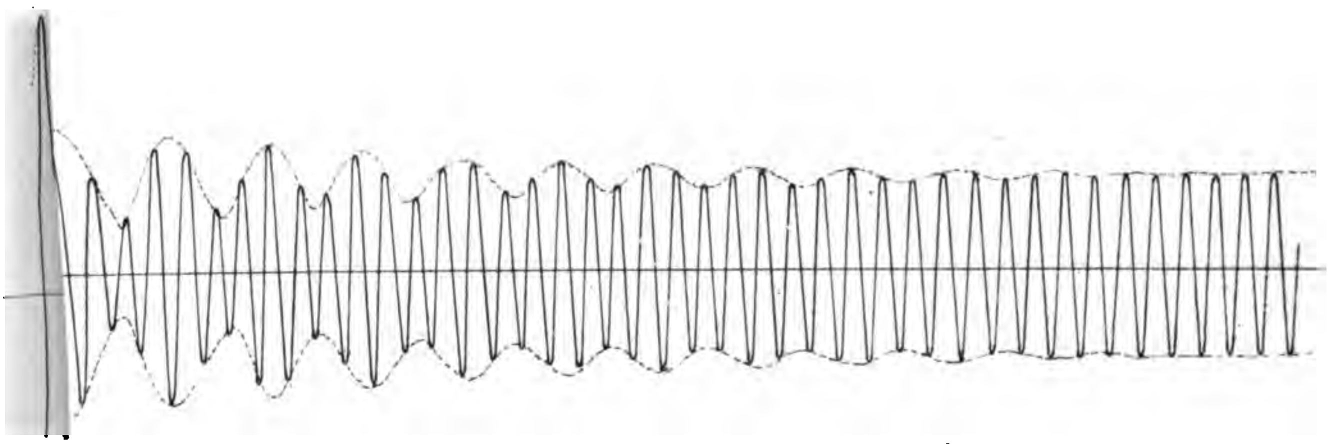


FIG. 2.

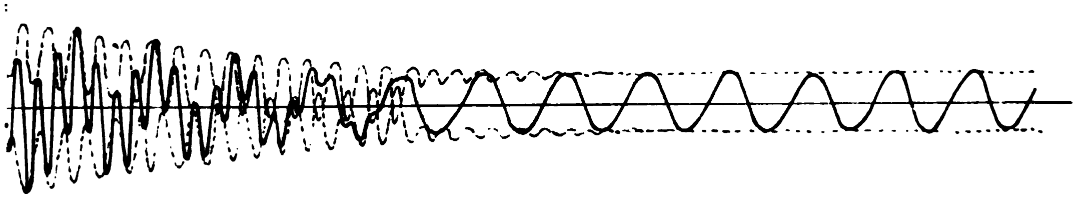


FIG. 4.

To Illustrate Captain G. Russo's

FIG. 1.

CURVE OF STABIL

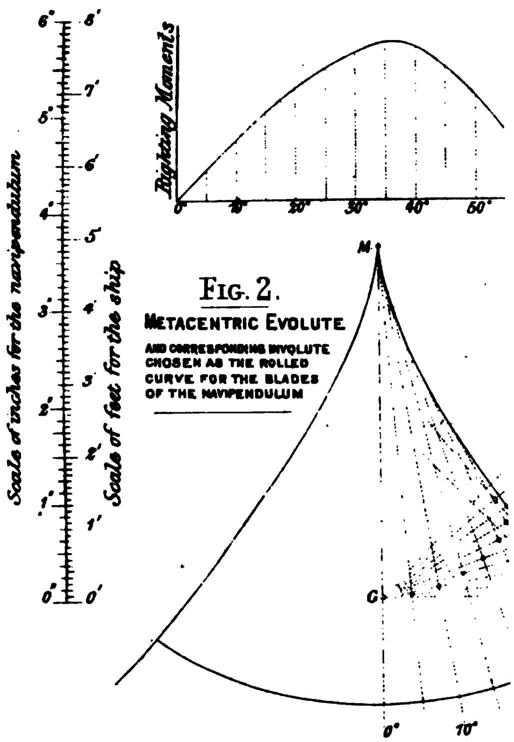
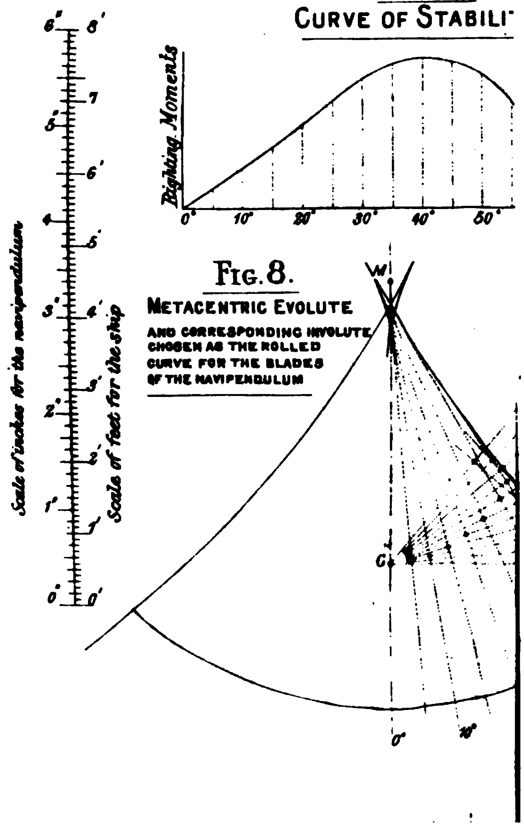


FIG. 7.

CURVE OF STABILI



To Illustrate Captain G. Russo's Paper on "The Navipendular Method of Experiments, as applied to some Warships of different Classes."

FIG. 1.

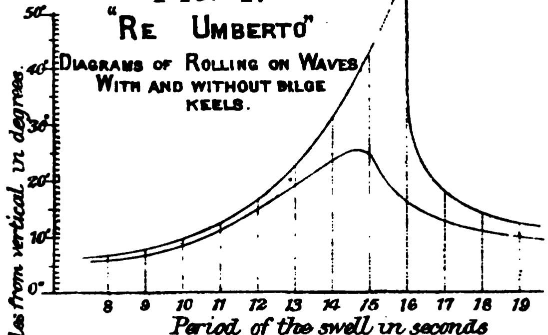


FIG. 2.

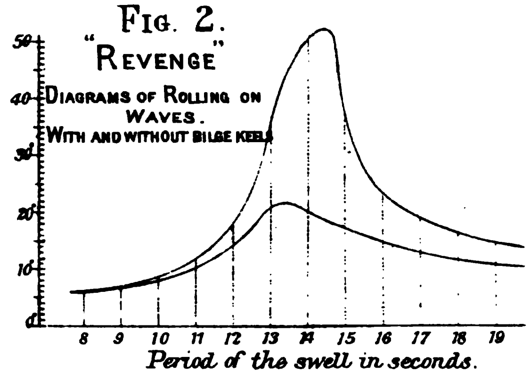


FIG. 3.

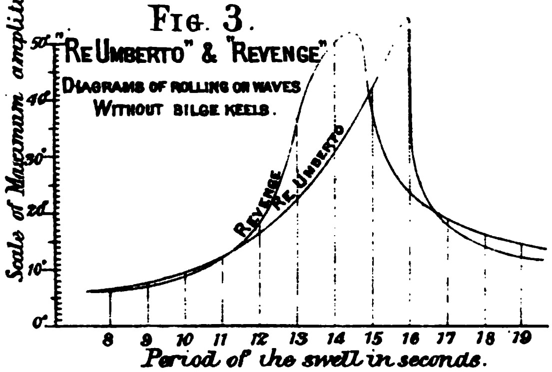


FIG. 4.

"RE UMBERTO", "REGINA MARGHERITA" "REVENGE"
DIAGRAMS OF ROLLING ON WAVES.
WITH ACTUAL BILGE KEELS.

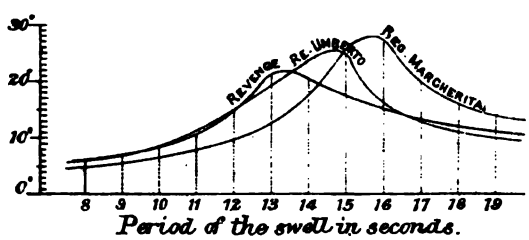
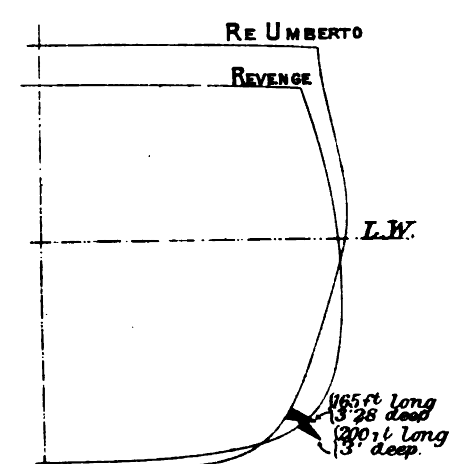


FIG. 5.

"RE UMBERTO" AND "REVENGE"
MIDSHIP SECTIONS.
SHOWING SIZE AND POSITION OF BILGE KEELS.



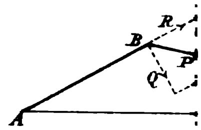


FIG. 1

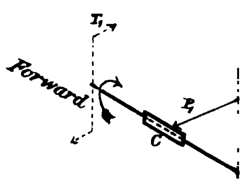


FIG. 6

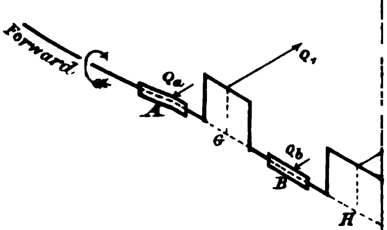


FIG. 9

ctual (

ctual Case of a Four-cranked Marine Shaft."

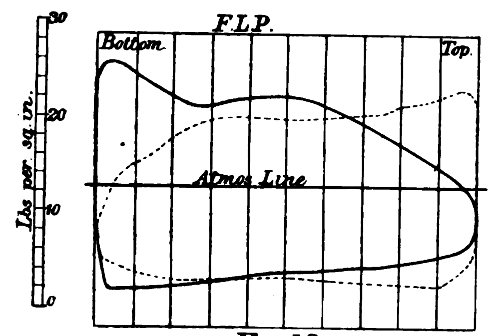


Fig. 18.

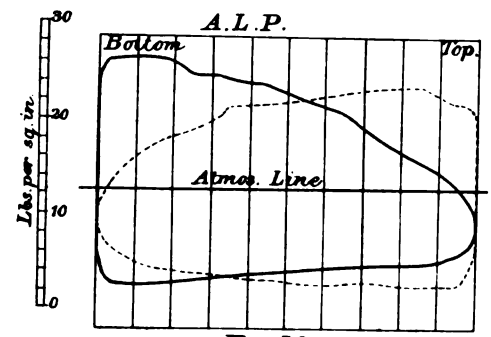


Fig. 19.

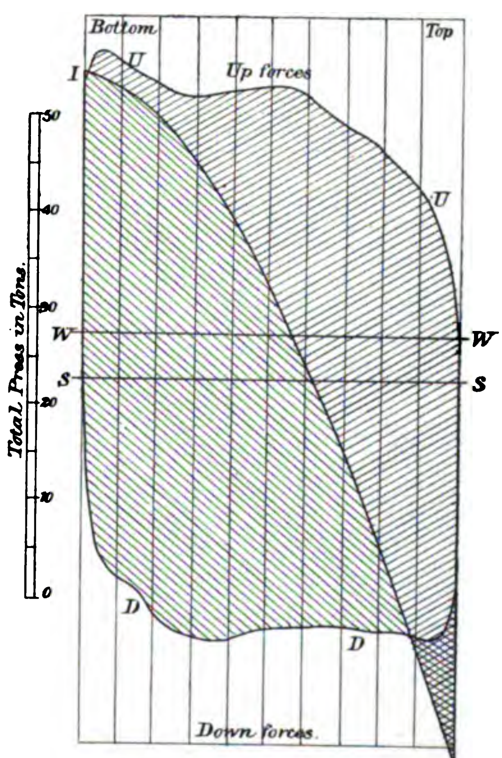


Fig. 22.

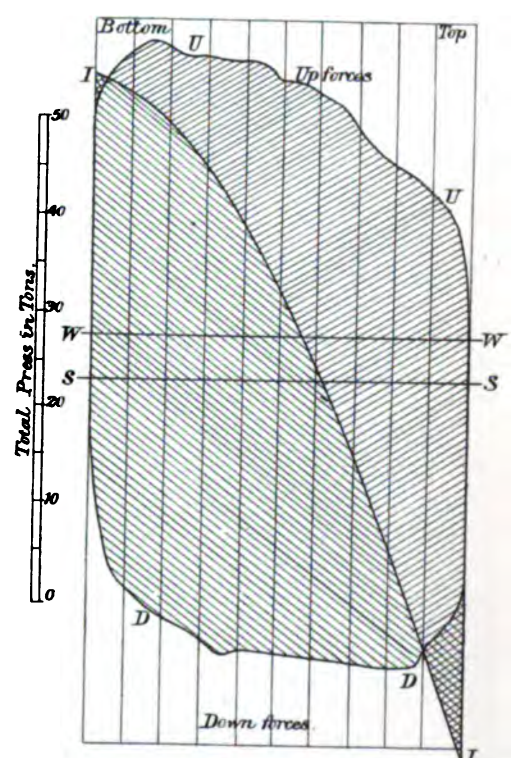
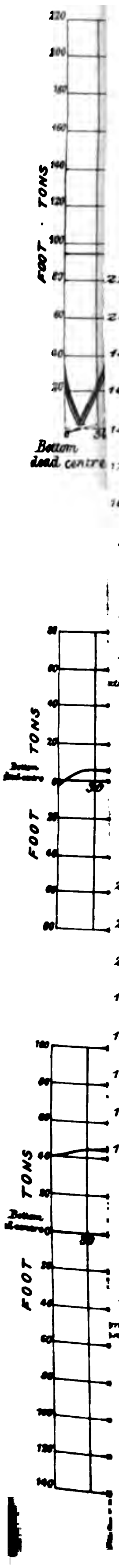


Fig. 23.



Twisting Moment on Pin.

M.P.

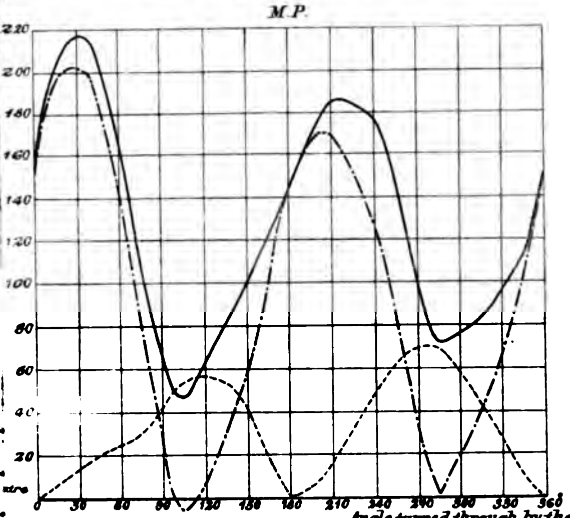


Fig. 33.

Equivalent Twisting Moments on Pin

M.P.

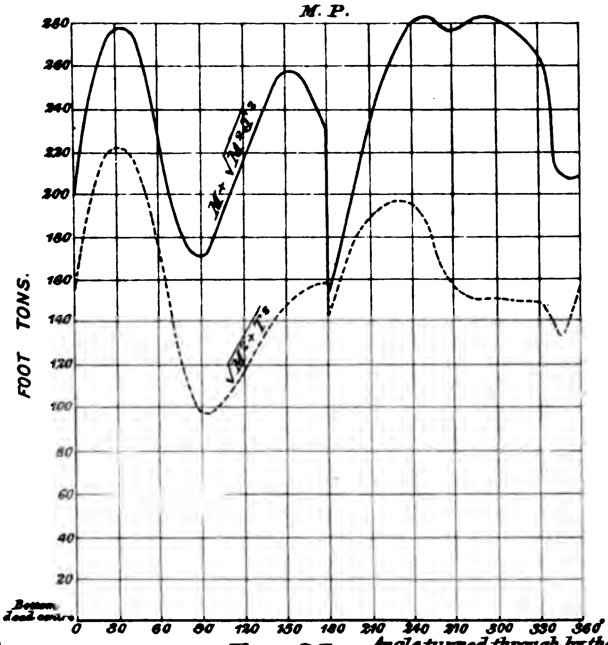


Fig. 37.

Twisting Moment on Pin.

A.L.P.

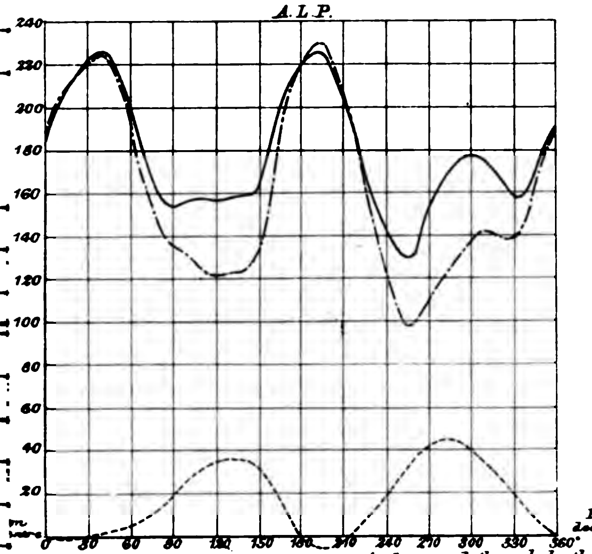


Fig. 34.

Equivalent Twisting Moments on Pin

A.L.P.

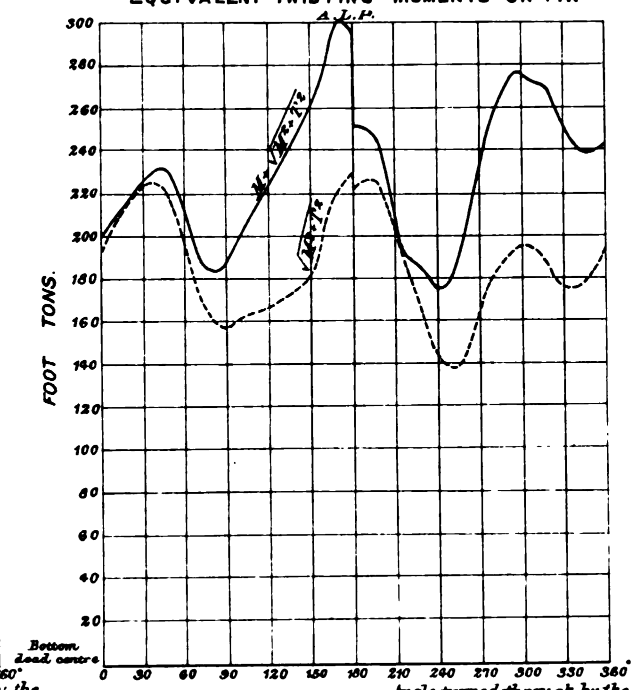


Fig. 38.

Parts of a

''

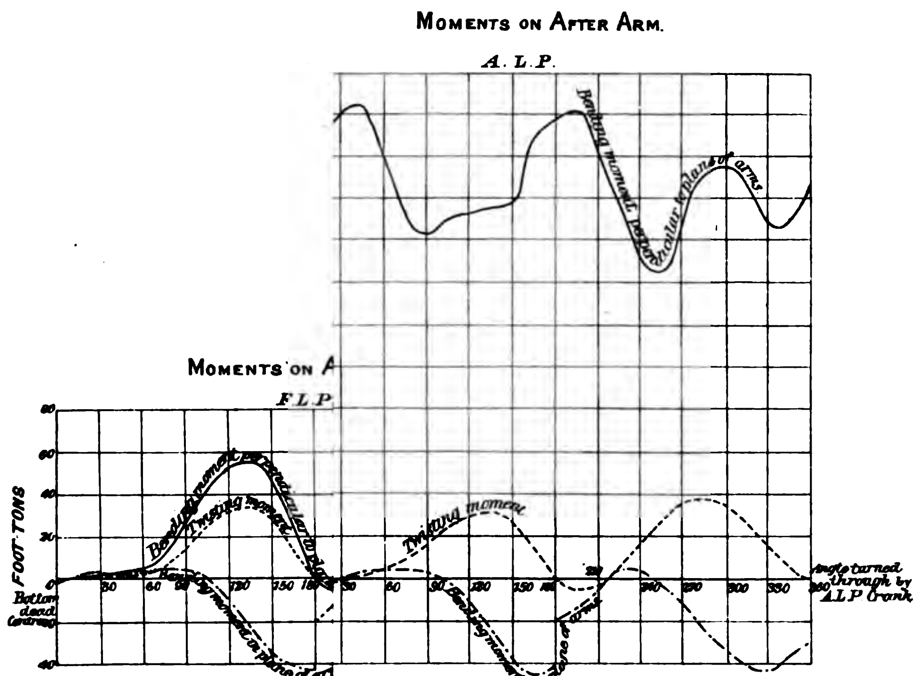


FIG. FIG. 42.

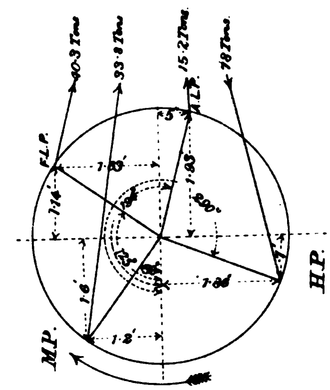


FIG. 44.

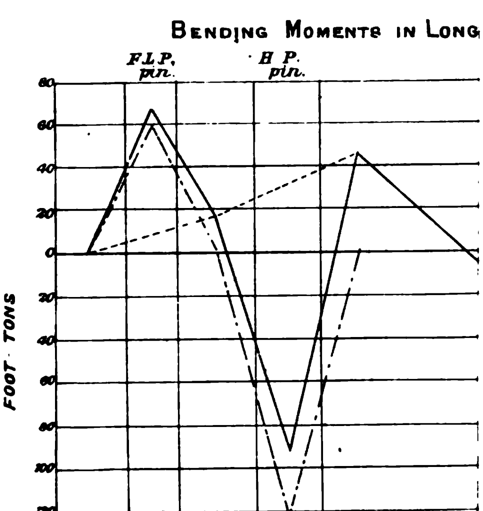


FIG. 45

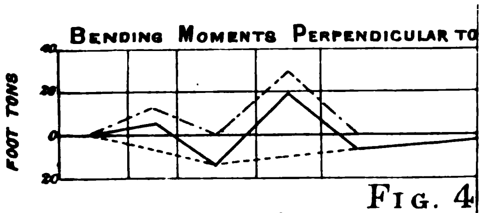
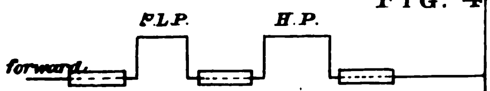


FIG. 4



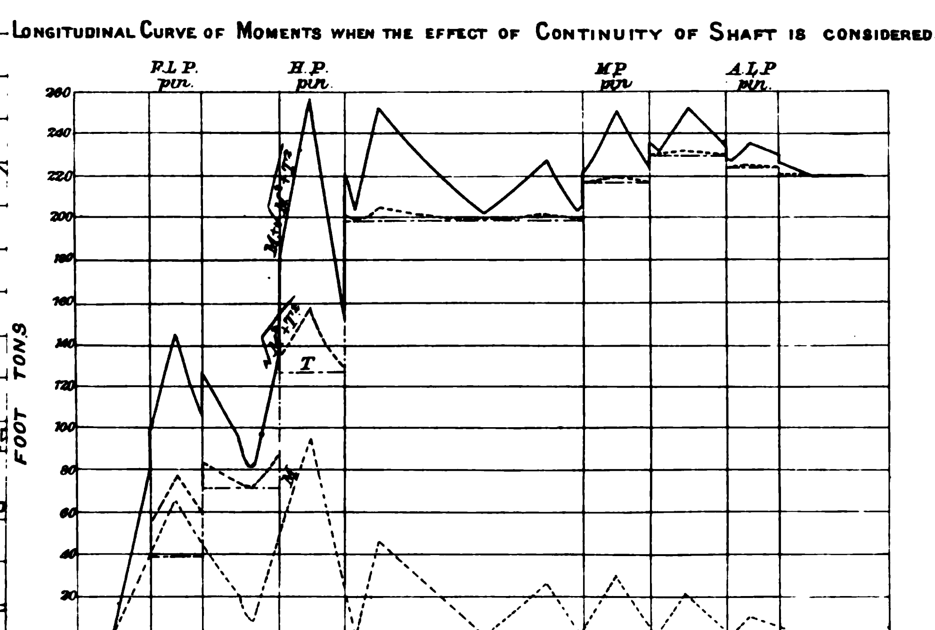
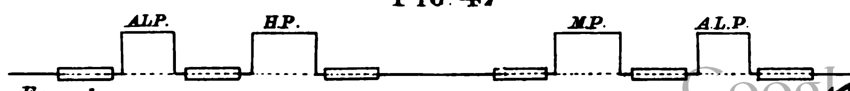


FIG. 47



To Illustrate Professor S. Dunkerley's Paper, "The Straining Actions on the different Parts of a Crank Shaft, illustrated by an actual Case of a Four-cranked Marine Shaft."

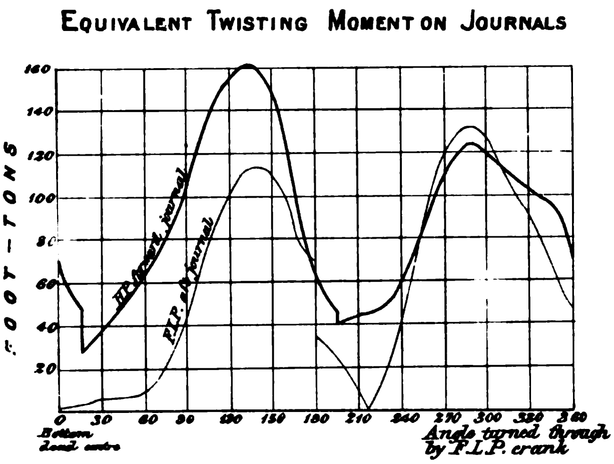


FIG. 48.

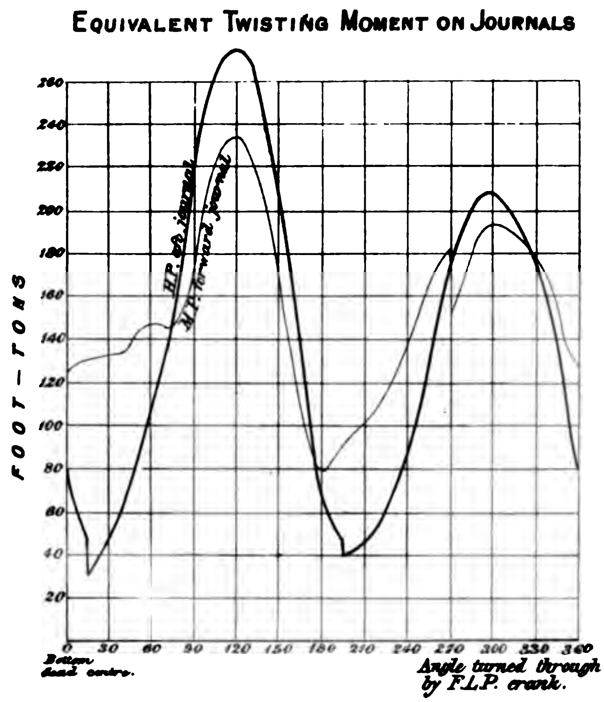


FIG. 49.

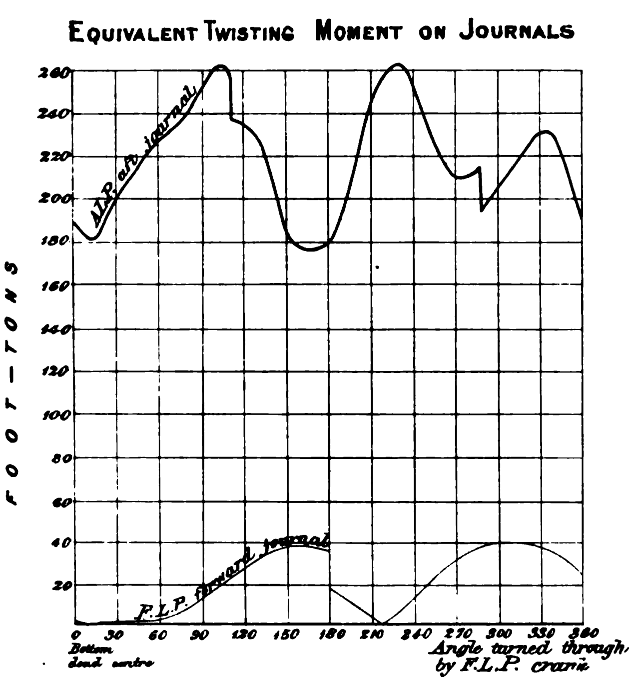


FIG. 51.

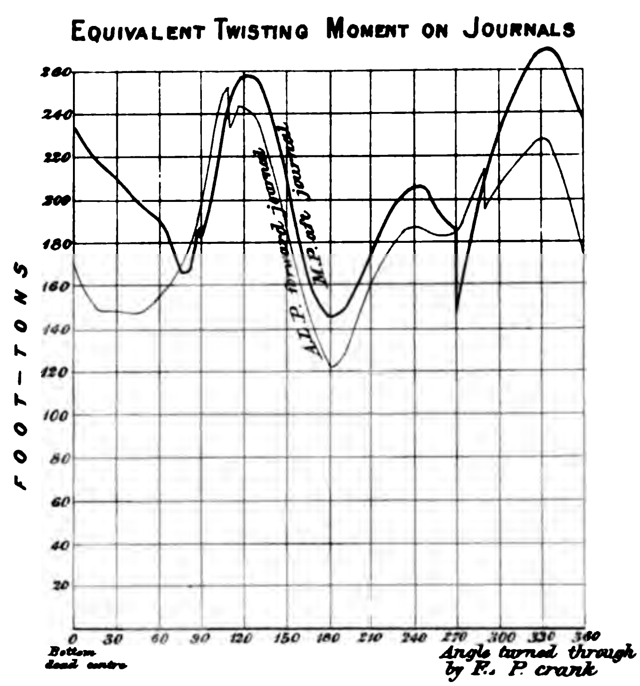
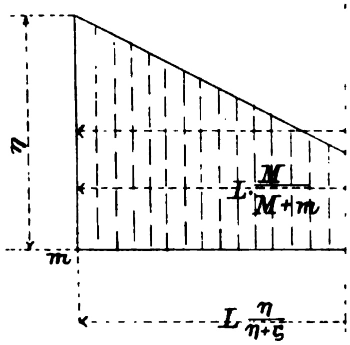


FIG. 50.


η
L·M/(M+m)
m
L η/(η+ζ)

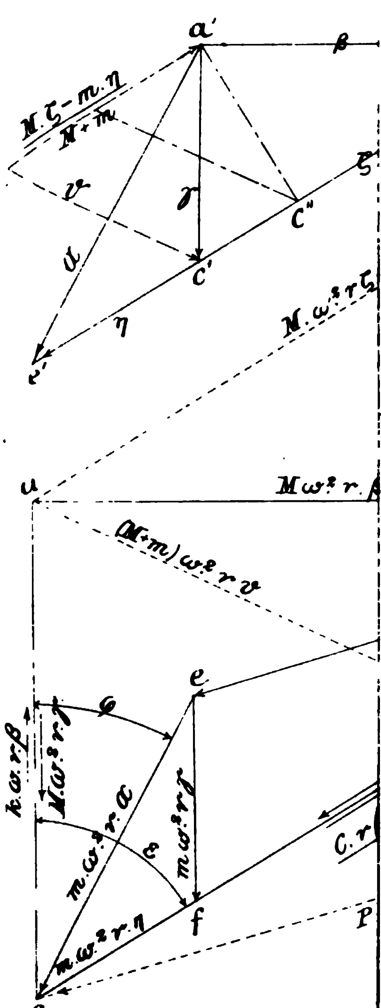

a′
β
(M.ζ − m.η)/(M+m)
ζ
v
γ
C″
u
c′
η
M.ω².r.ζ
u
Mω². r. β
(M+m) ω². r. v
e
φ
k.ω.r.β
M.ω².r.γ
m.ω².r.α
ε
m.ω².r.γ
m.ω².r.η
f
C.r
P
c

FIG. 3.

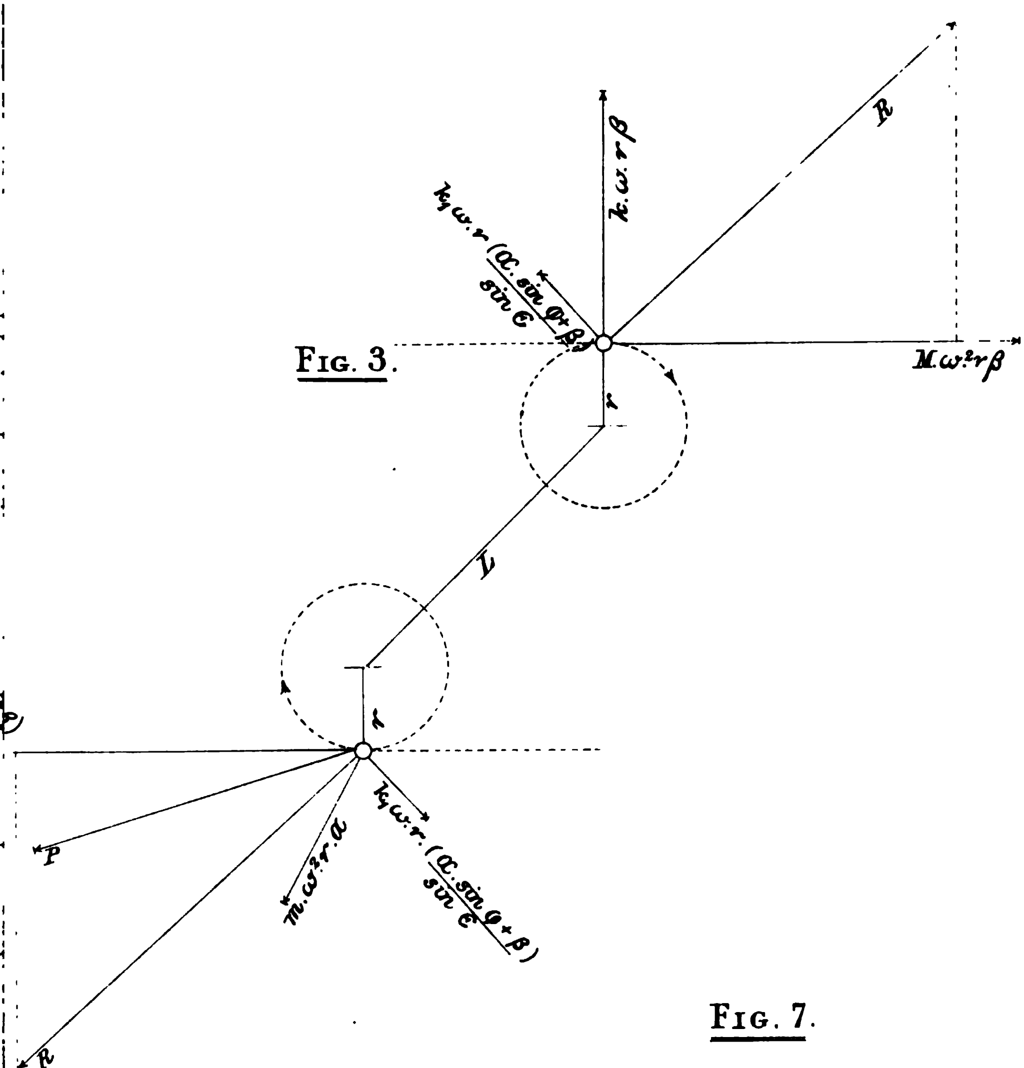
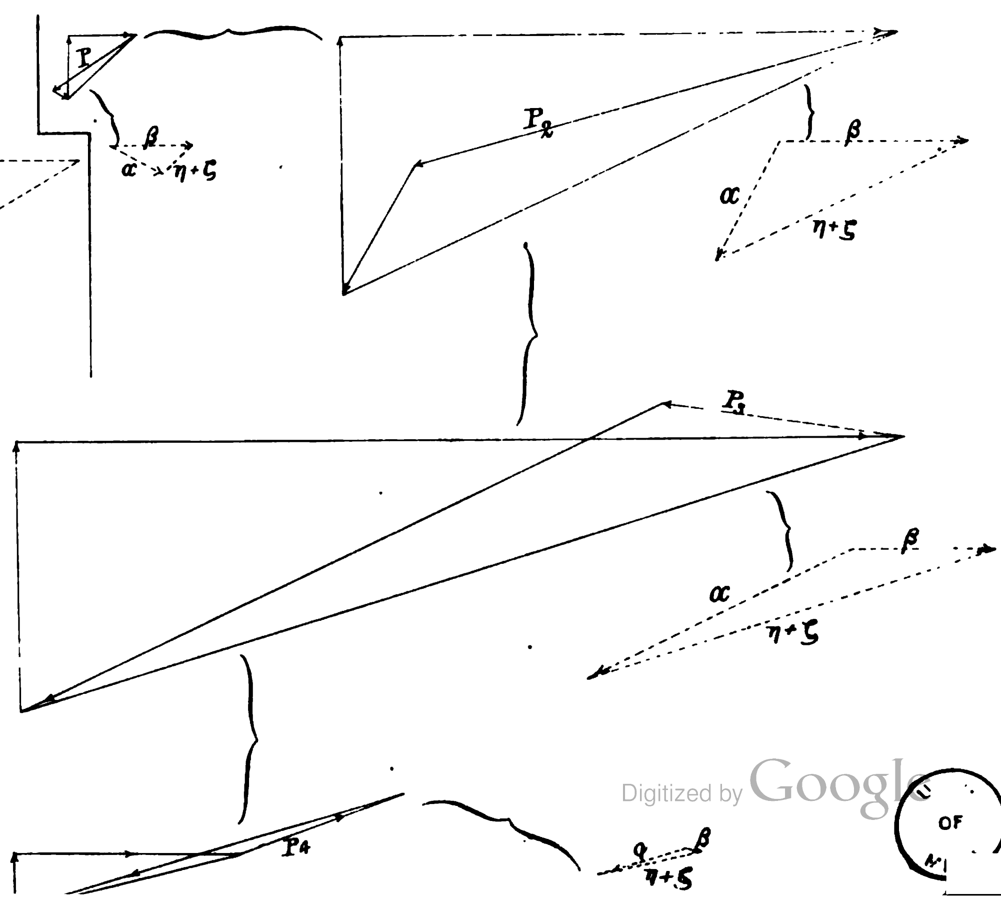
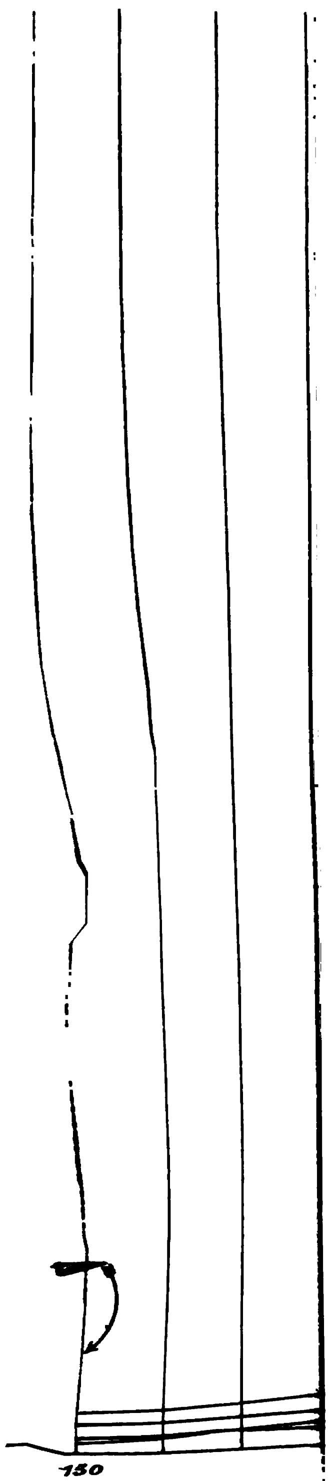
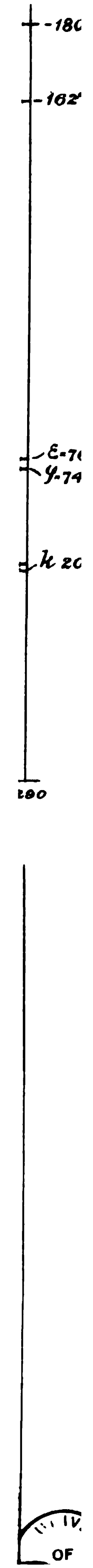

R
k.ω.r.β
k₁ω.r.(α.sin φ+β)/sin ε
Mω²rβ
r
L
r
P
m.ω².r.α
k₁ω.r.(α.sin φ+β)/sin ε
R

FIG. 7.


P
β
α
η+ζ
P₂
β
α
η+ζ
P₃
α
β
η+ζ
P₄
α
β
η+ζ

I.





To Illustrate Herr L. Gümbel's Paper, "Torsional Vibrations of Shafts."

FIG. 9

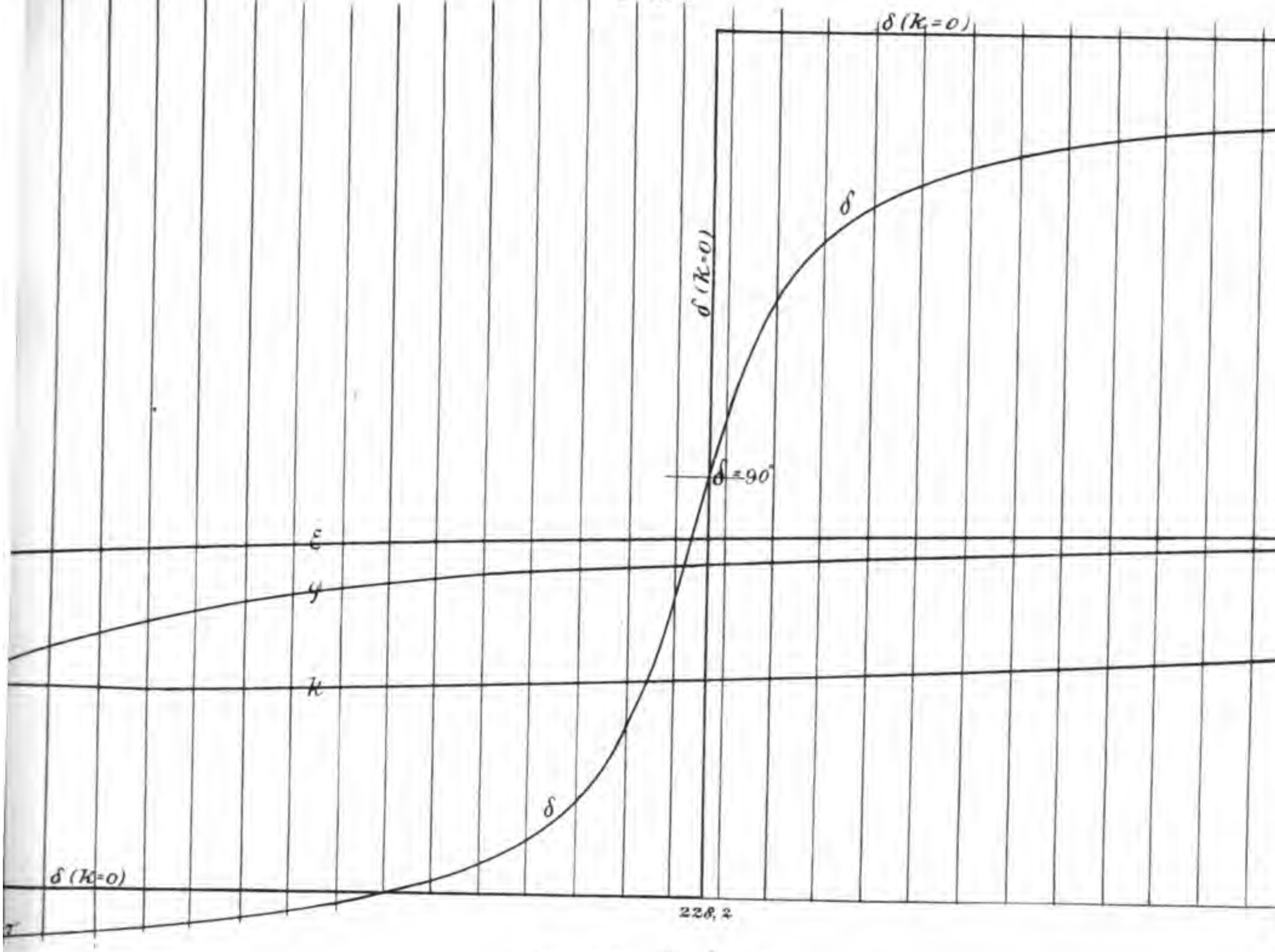
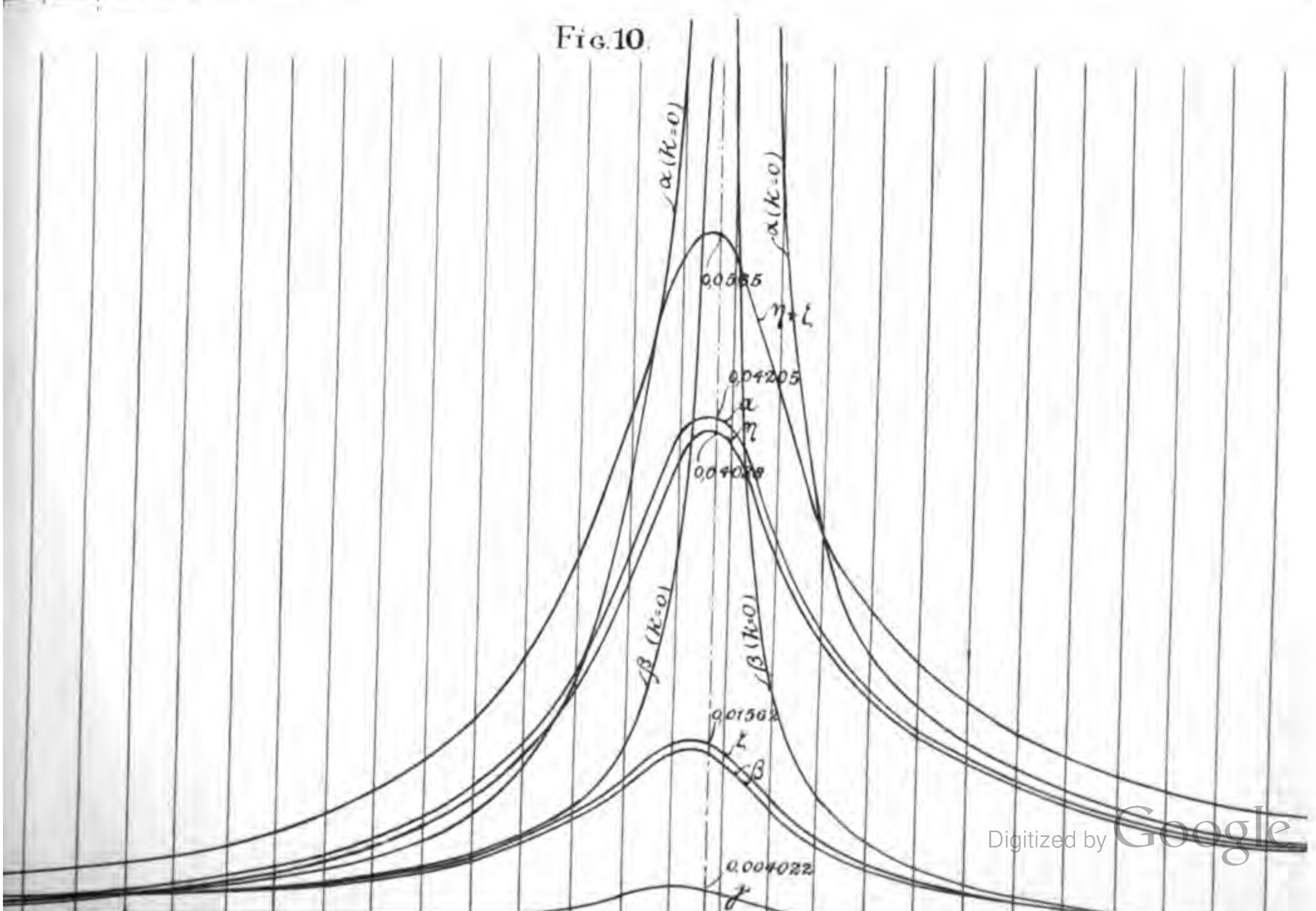
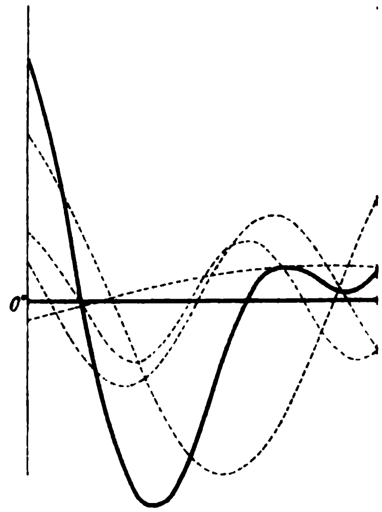


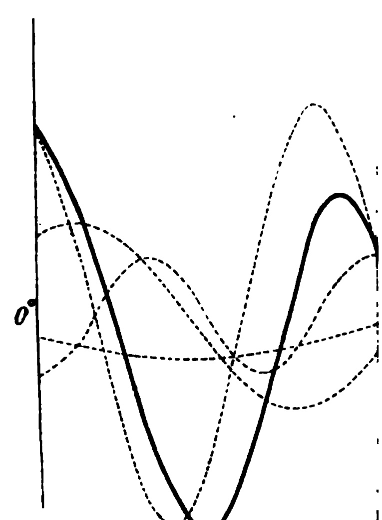
FIG. 10

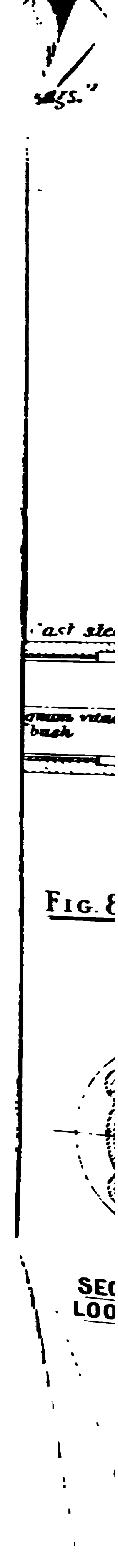


$P = 34760 + 1400 \cdot \cos(240 + X) + \ldots$
$+ 2410 \cdot \cos 4 (0 + X) (\ldots$



$\alpha = 0,127 + 0,00696 \cos (120^\circ \ldots$
$+ 0,03375 \cos 3 (8^\circ 4 \ldots$





…ngs."

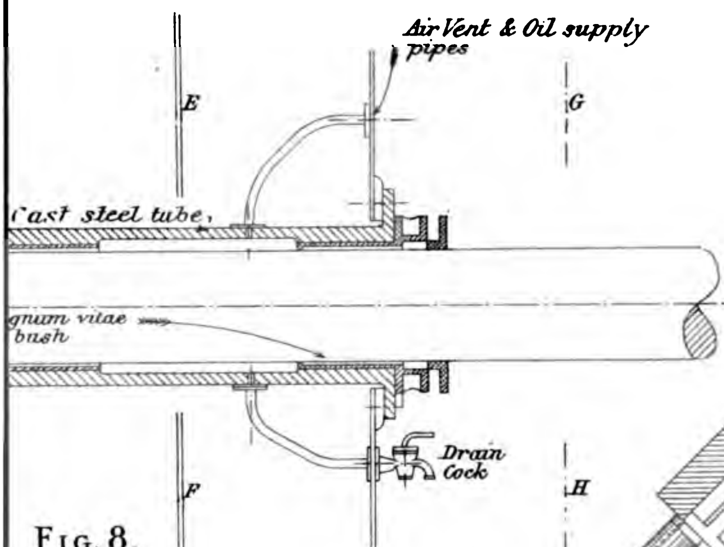
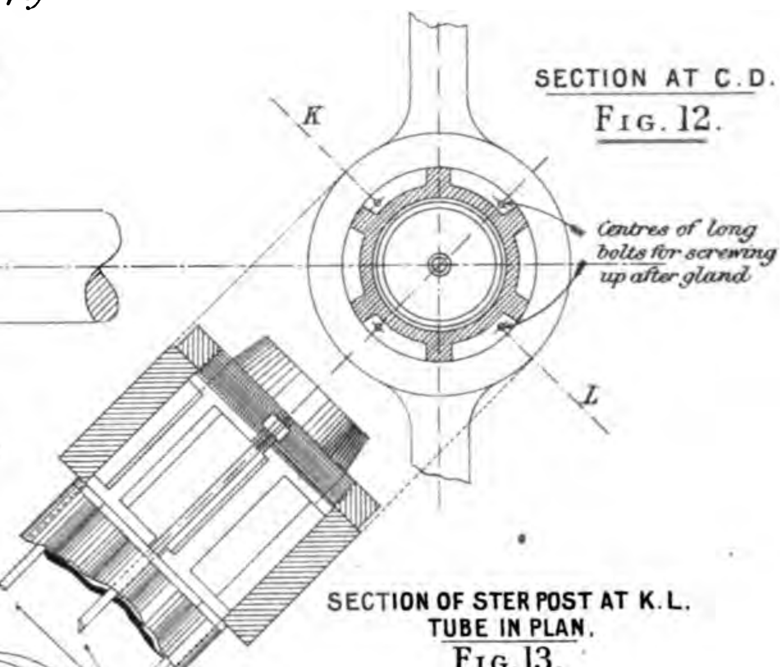
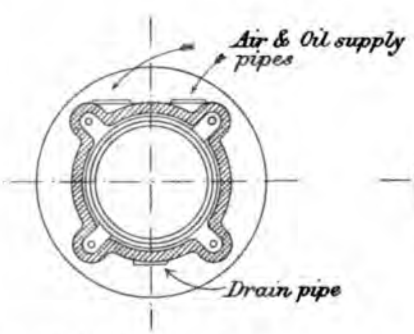


FIG. 8.



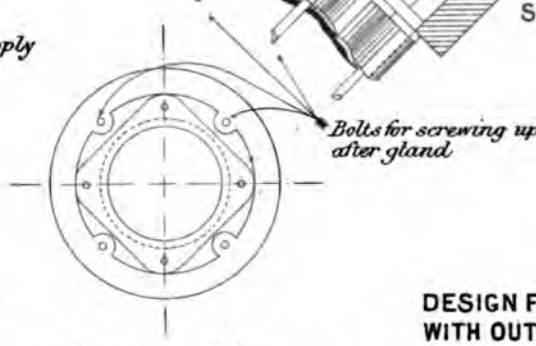
SECTION AT C.D.
FIG. 12.

SECTION OF STER POST AT K.L.
TUBE IN PLAN.
FIG. 13.



SECTION AT E.F.
LOOKING FORWARD

FIG. 10.



END VIEW AT G.H.

FIG. 11.

DESIGN FOR CAST STEEL STERN TUBE
WITH OUTER GLAND ADJUSTABLE FROM
INSIDE TUNNEL.
YOUNGER AND KINGS PATENT.

Shaft Bearings."

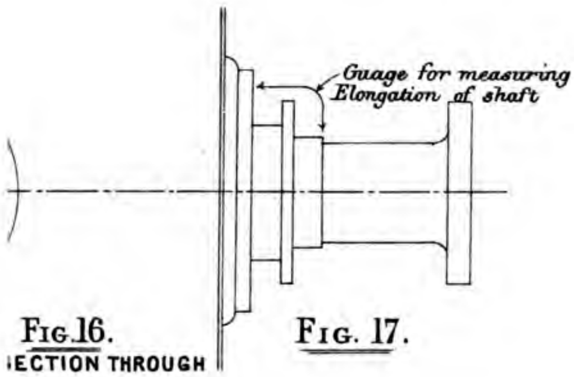


Fig. 16.
ECTION THROUGH
STERN POST.

Fig. 17.

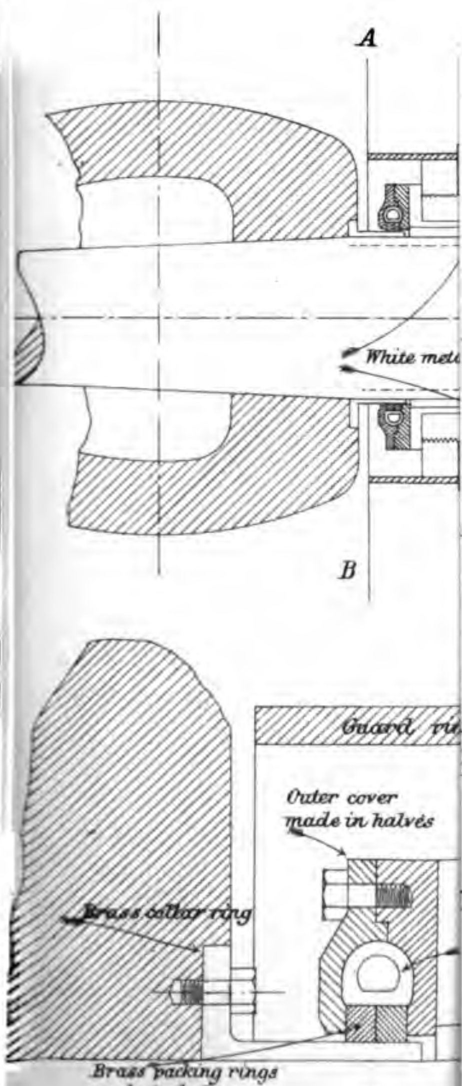
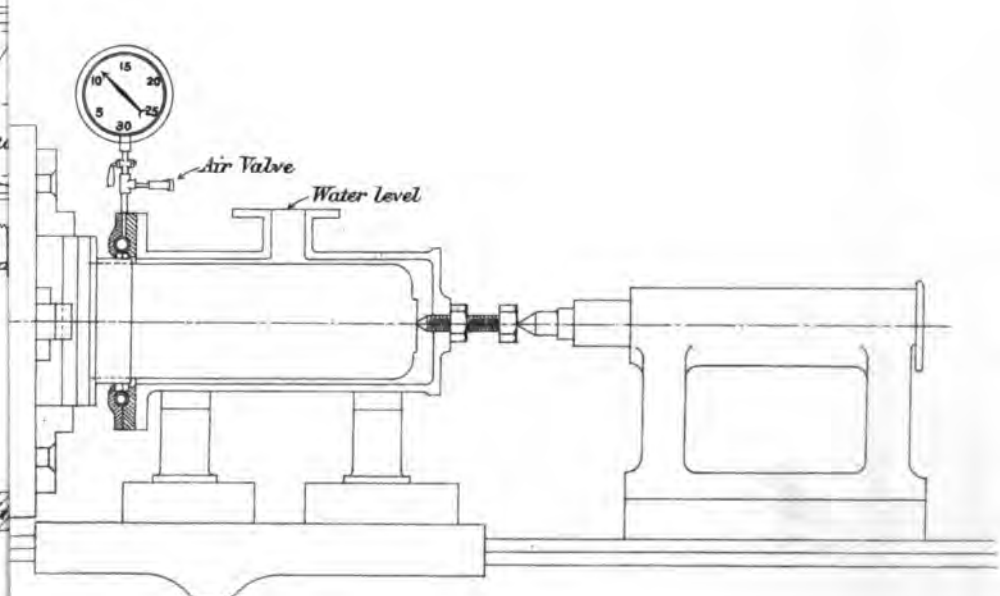


Fig. 2



H OF EXPERIMENTAL APPARATUS

Fig. 21.

To Illustrate Mr. C. E. Stromeyer's Paper, "Distortion of Boilers due to Overheating."

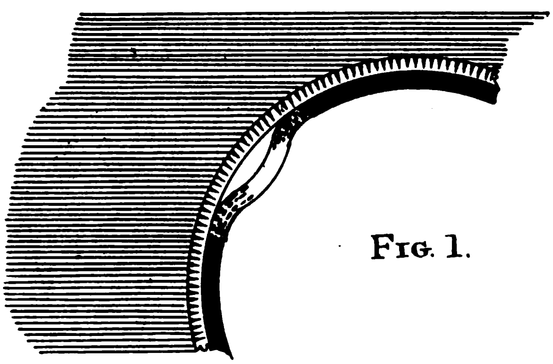
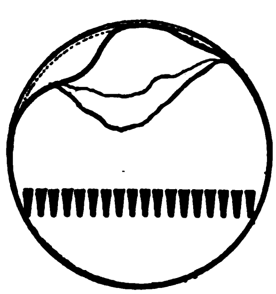


FIG. 1.

FIG. 2.



SECTION ON AB

FIG. 3.

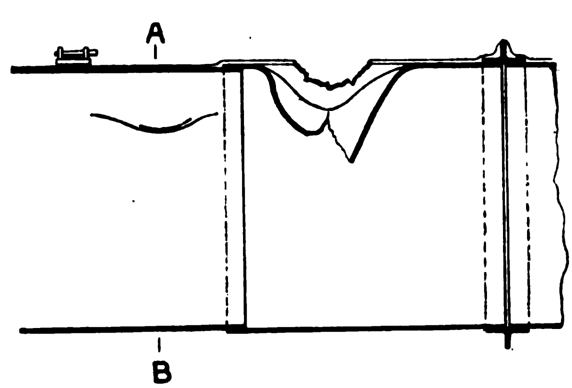


FIG 4

To Illustrate Mr. C. E. Ellis's remarks on Mr. C. E. Stromeyer's Paper, "Distortion in Boilers due to Overheating."

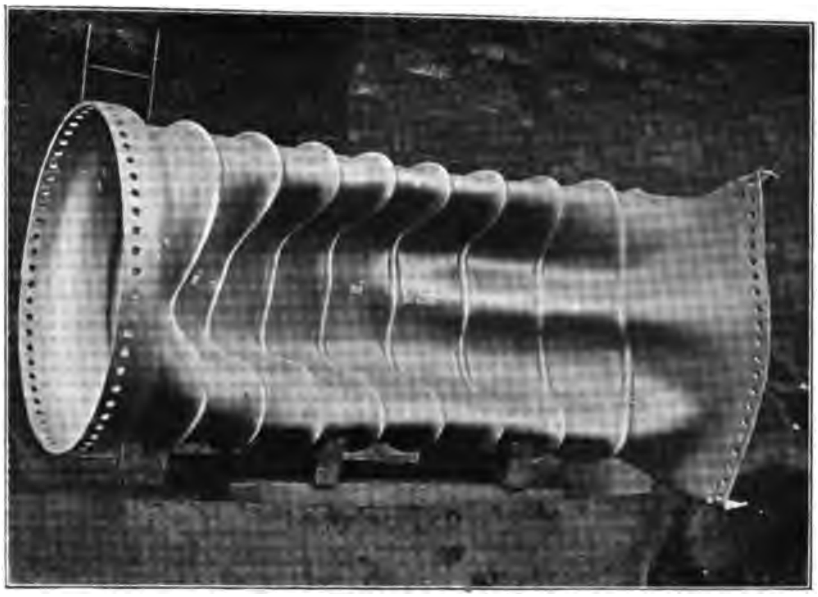


Fig. 1.

Fig. 2.

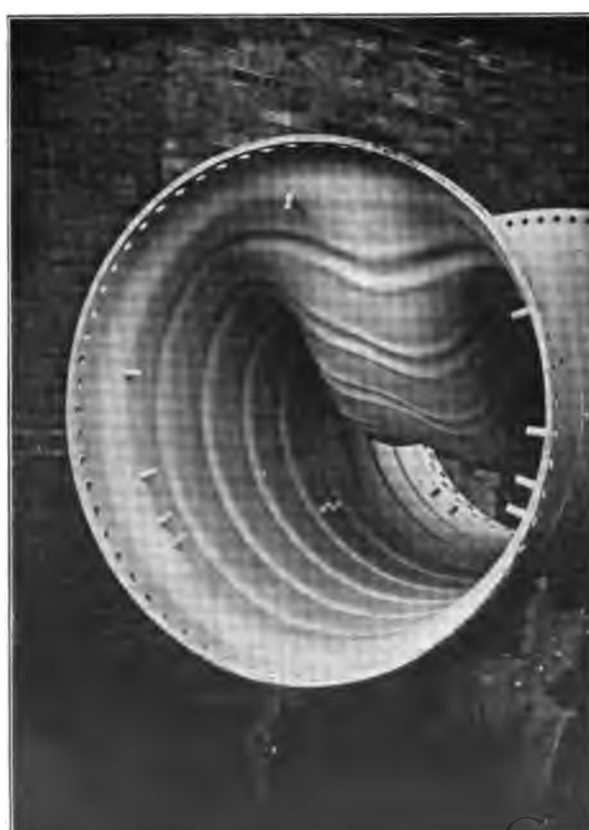
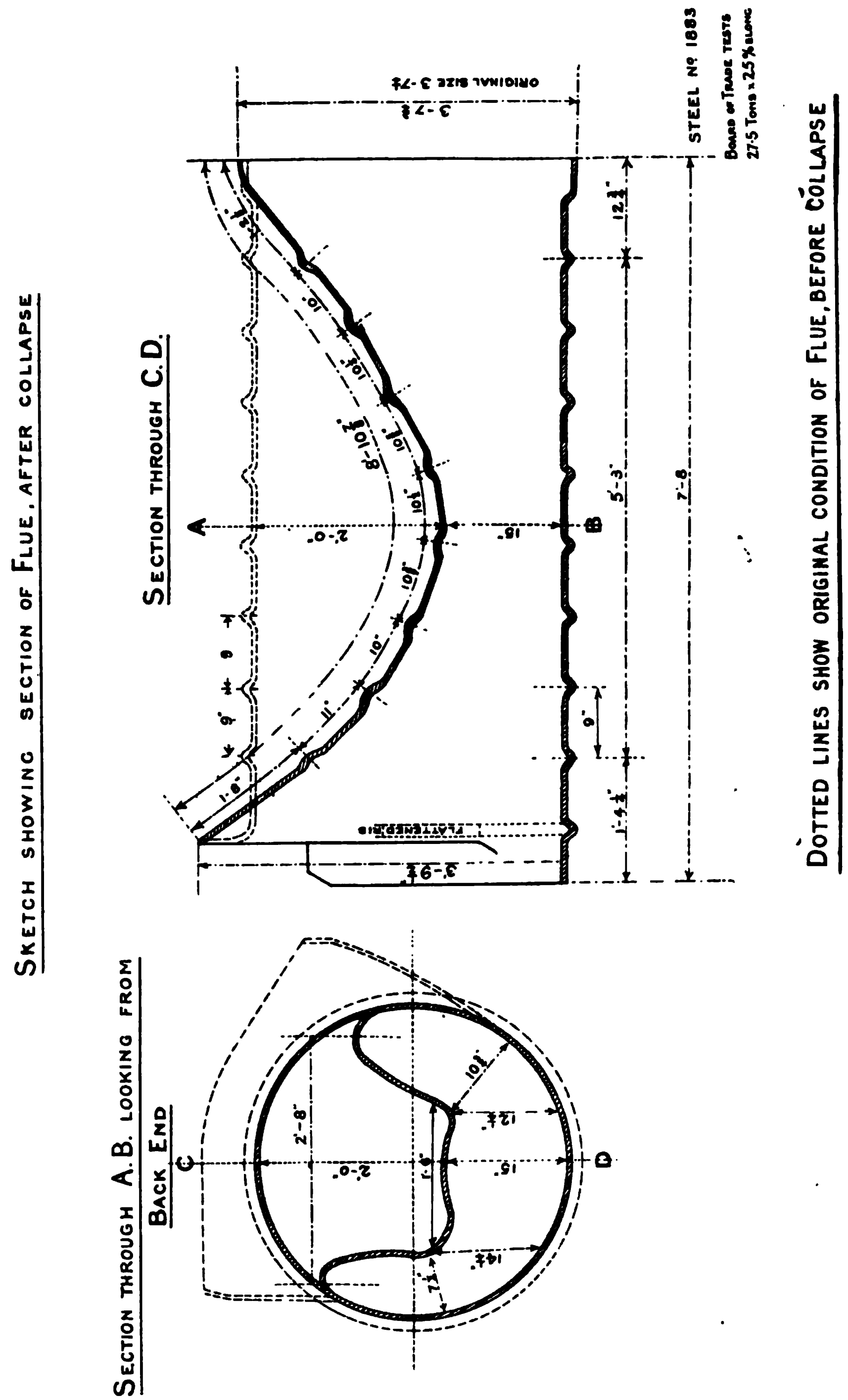


Fig. 3.

To Illustrate Mr. C. E. Ellis's remarks on Mr. C. E. Stromeyer's Paper,
"Distortion in Boilers due to Overheating."



To Illustrate Professor Ernst Cohen's Paper on "The Corrosion of Condenser Tubes and Sea-Water Conductors."

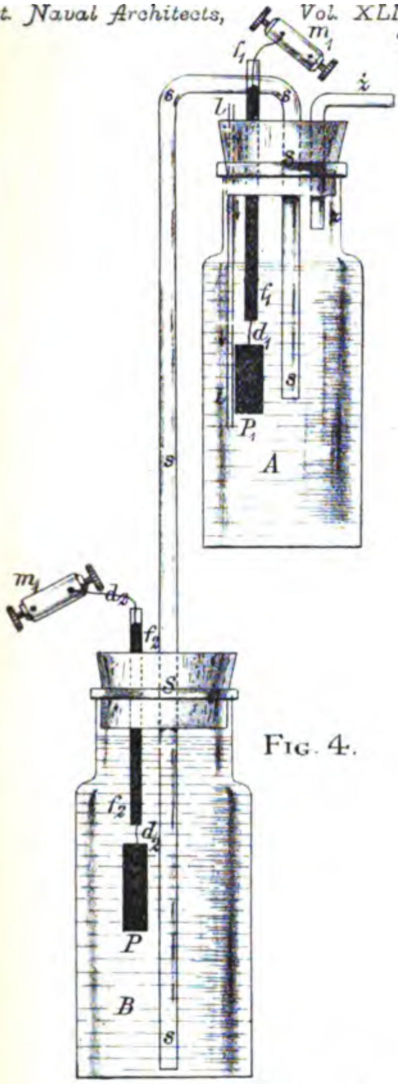


FIG. 4.

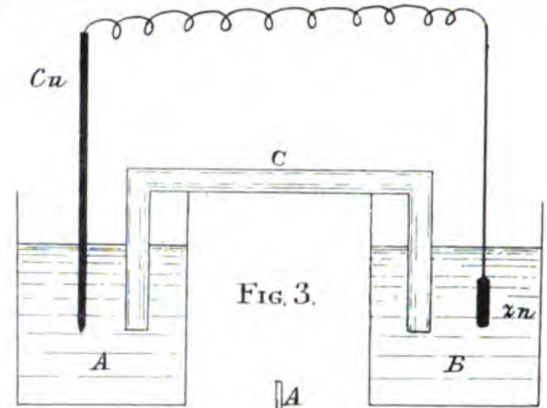


FIG. 3.

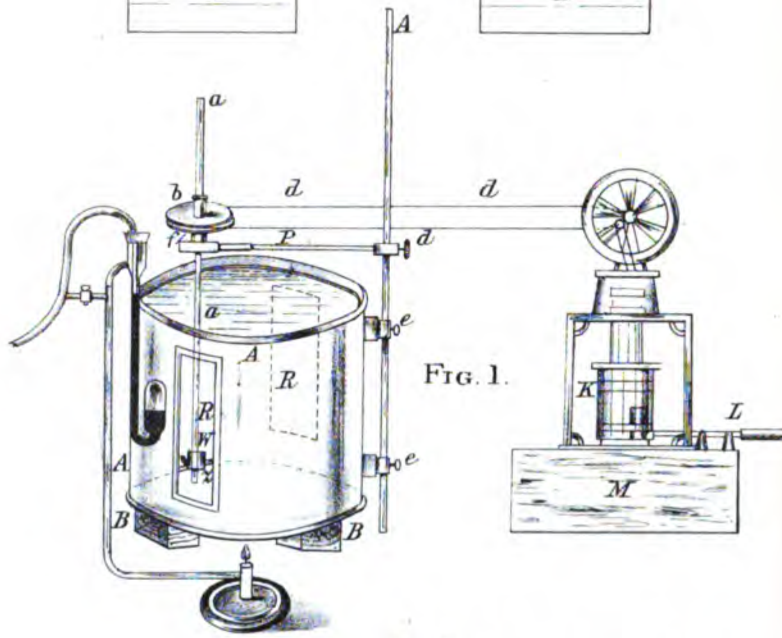


FIG. 1.

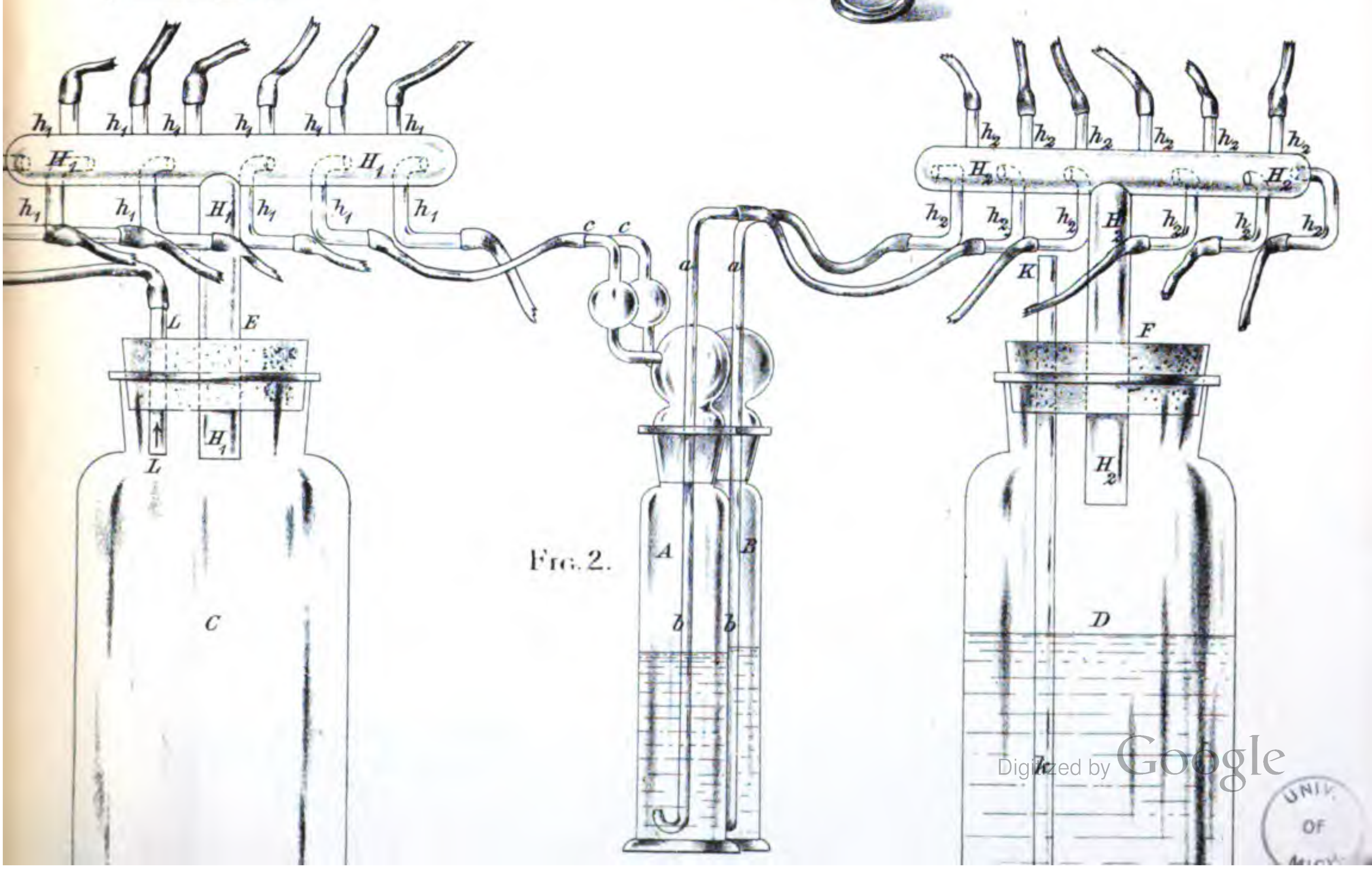
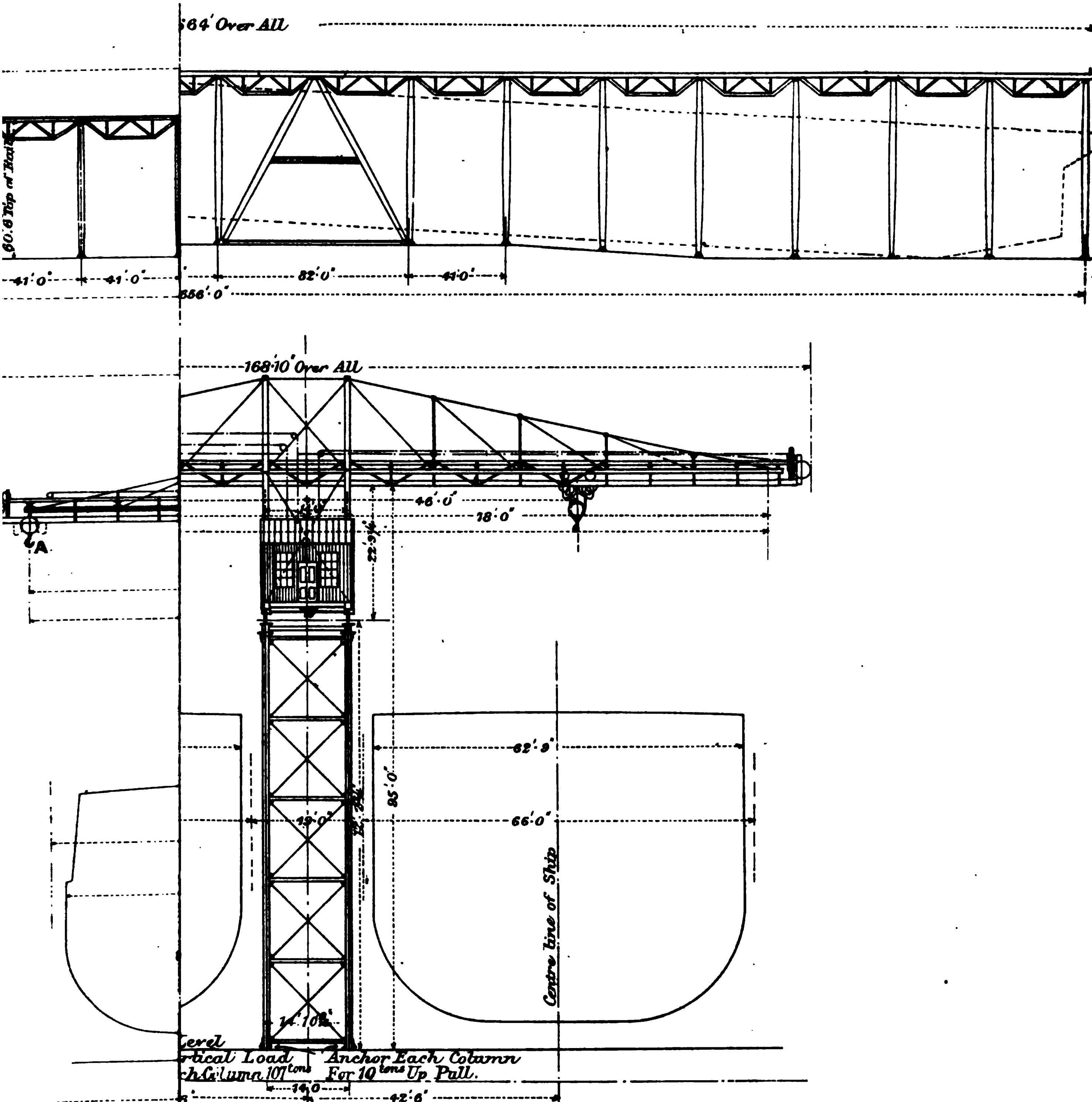
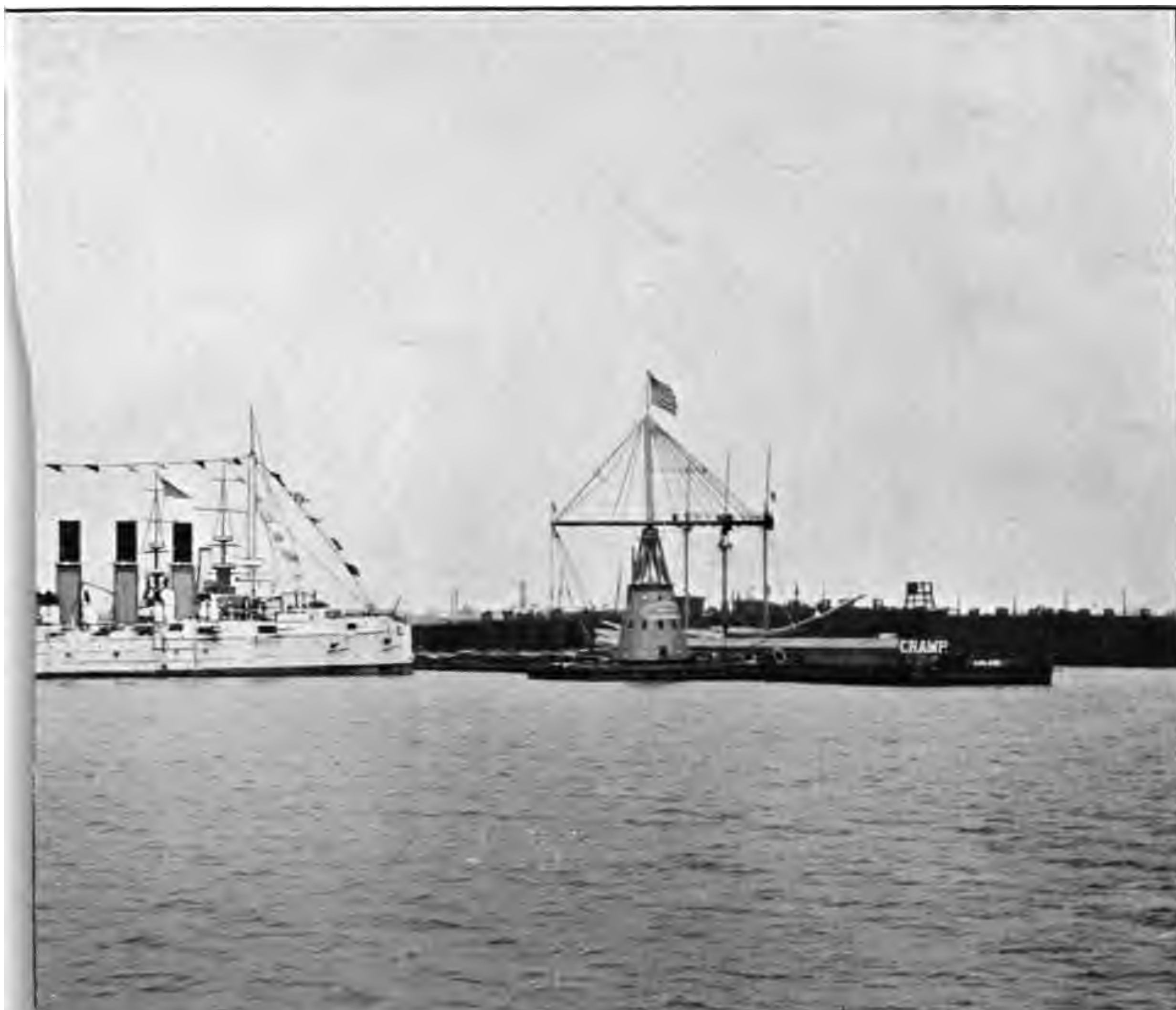


FIG. 2.

in American Shipyards."



OWN CANTILEVER CRANE.
Wm. CRAMP & SONS.
FIG 2


CRAMP

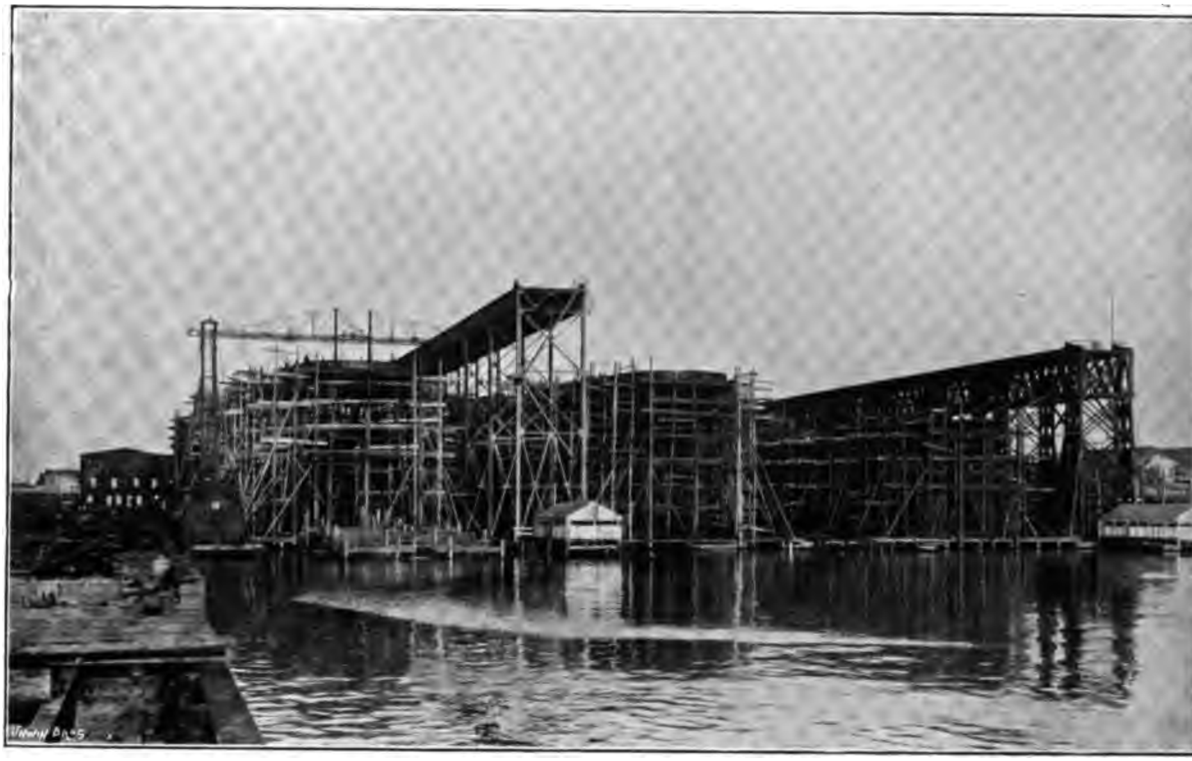


Fig. 3.

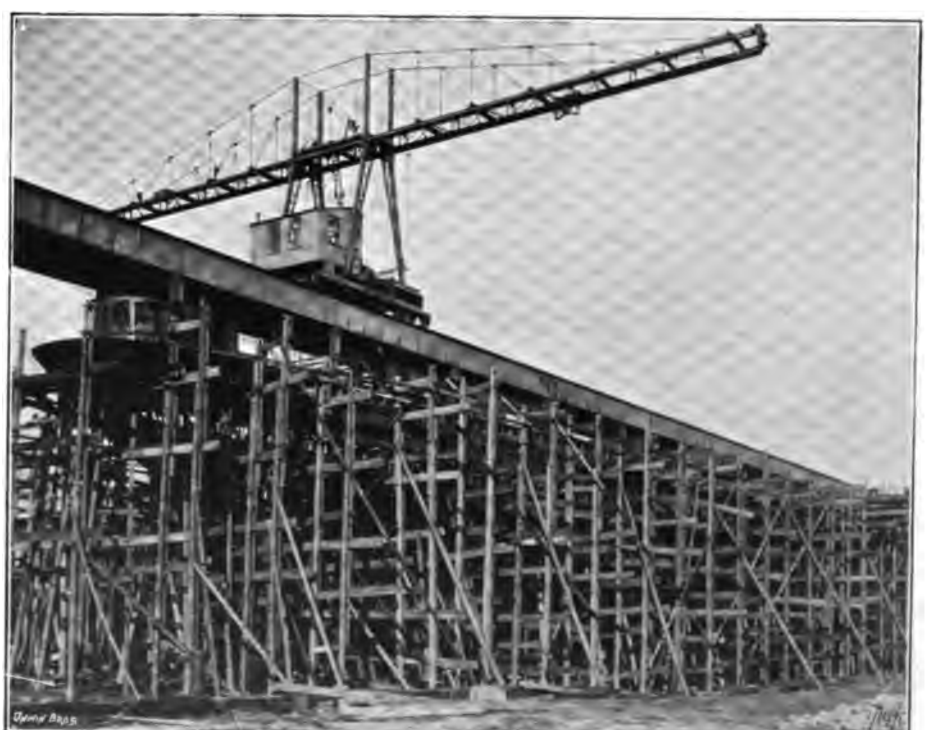
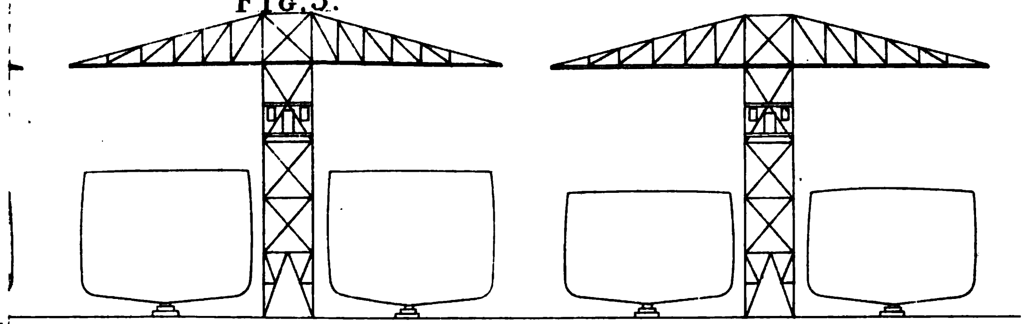


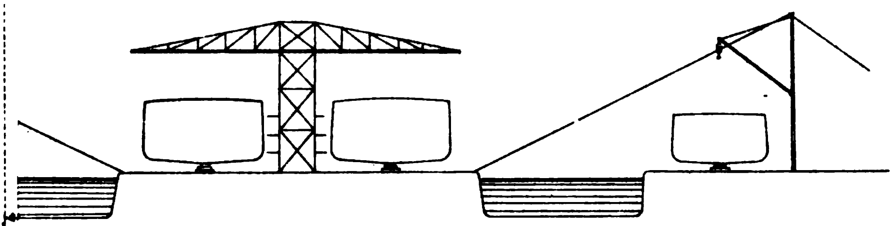
Fig. 6.

…TCH SHOWING GENERAL LAY OUT OF …G BERTHS AND CRANE SERVICE OVER SAME … VARIOUS AMERICAN SHIPYARDS.

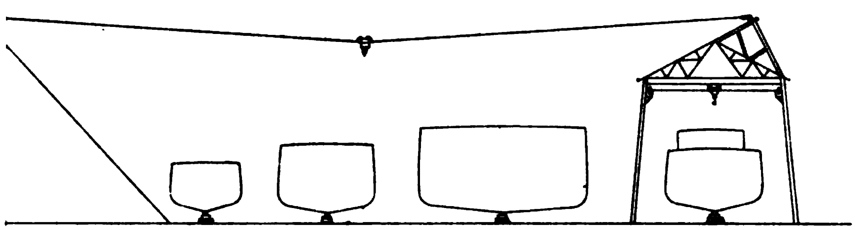
FIG. 3.



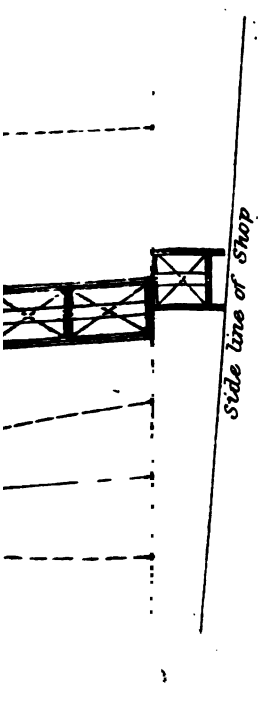
C… …ILEVER CRANES, NEWPORT NEWS, S & D. D. C°
STERN LAUNCHING.

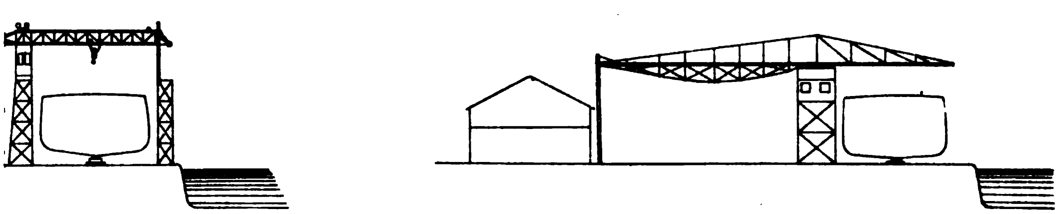


…ANTILEVER CRANE, DETROIT DRY DOCK C°
SIDE LAUNCHING.



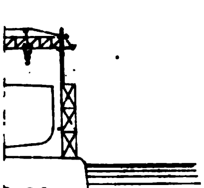
…RE ROPE TROLLEY & SHIP HOUSE, BATH IRON WORKS
STERN LAUNCHING



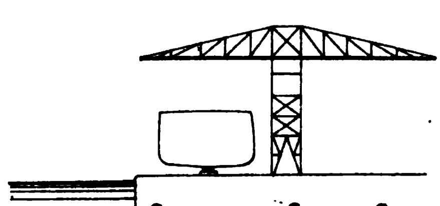


… S. B. C°
…G.

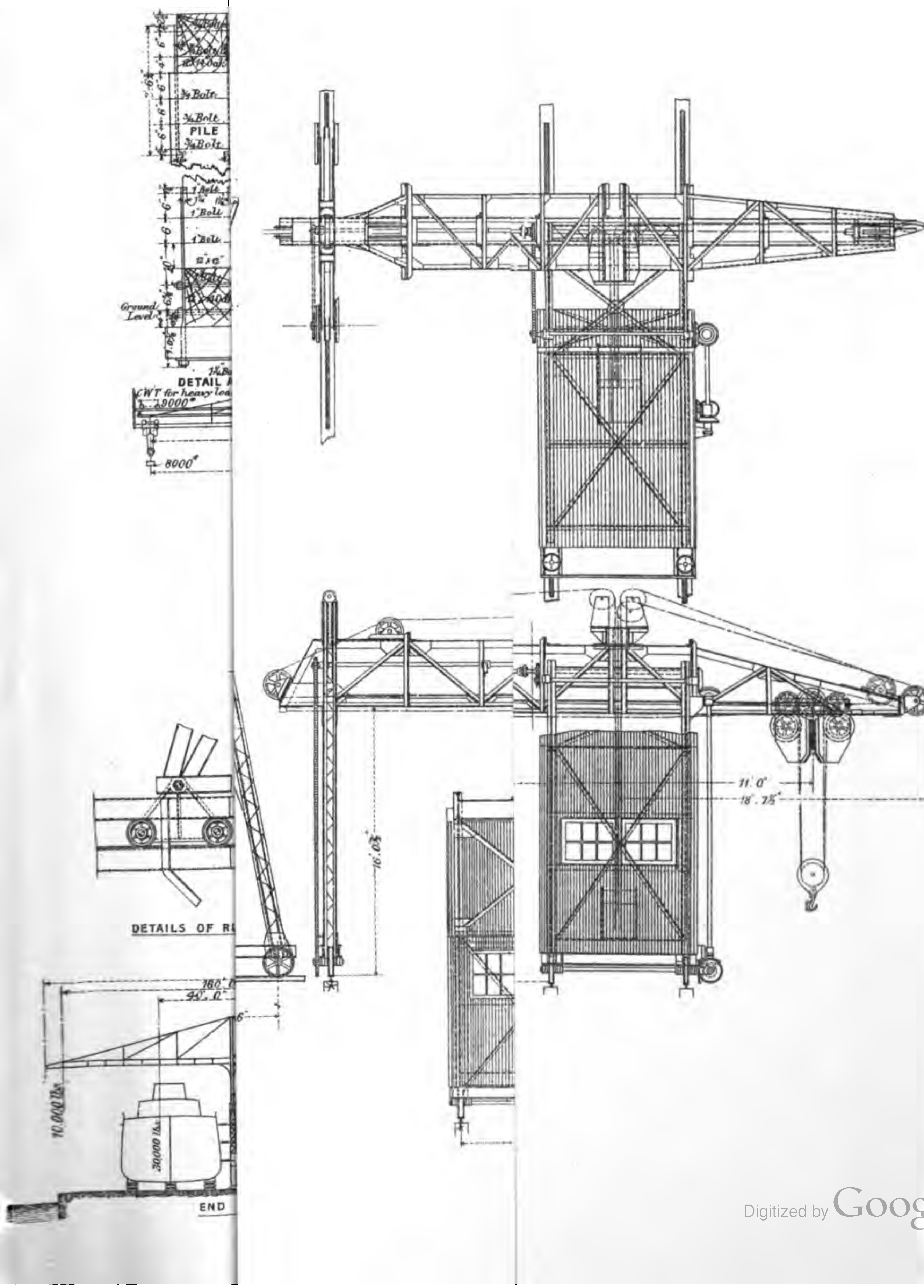
GANTRY CRANE, WITH CATILEVER EXTENSION
WEST BAY CITY, SIDE LAUNCHING.



…B. C°



CANTILEVER CRANE, CHICAGO S. B. C°
SIDE LAUNCHING.



¾ Bolt
¾ Bolt
PILE
¾ Bolt
1" Bolt
1" Bolt
1" Bolt
12"x6"
Ground Level
DETAIL
CWT for heavy loa
19000#
8000#
11' 0"
18' 7½"
16' 0¾"
DETAILS OF R
160' 0
95' 0"
10,000 lbs.
30000 lbs.
END

5-TON EL

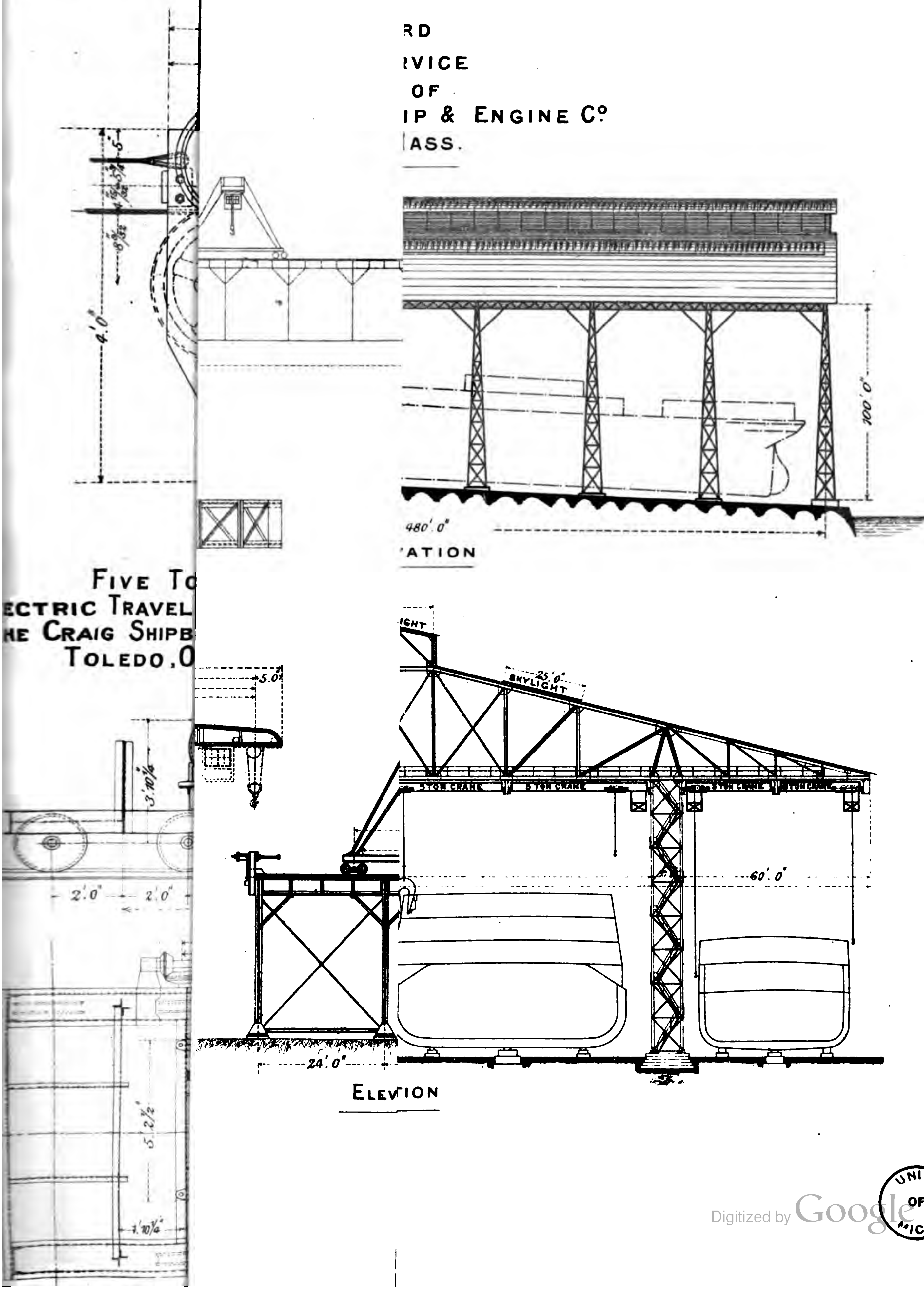

RD
RVICE
OF
IP & ENGINE Co
ASS.
100' 0"
480' 0"
ATION
FIVE TO
ECTRIC TRAVEL
HE CRAIG SHIPB
TOLEDO, O
4' 0"
8 9/32"
5"
5.0"
3' 10 1/4"
2' 0"
2' 0"
5' 2 1/2"
1' 10 1/4"
IGHT
25' 0"
SKYLIGHT
5 TON CRANE
5 TON CRANE
5 TON CRANE
5 TON CRANE
60' 0"
24' 0"
ELEVTION

Fig. 4.
BROWN GANTRY STEAM CRANE. (AMERICAN S.B. CO., WEST BAY CITY, MICH.)

To Illustrate Building Berths in American Shipyards."

Plate XXXII.

(DETAI

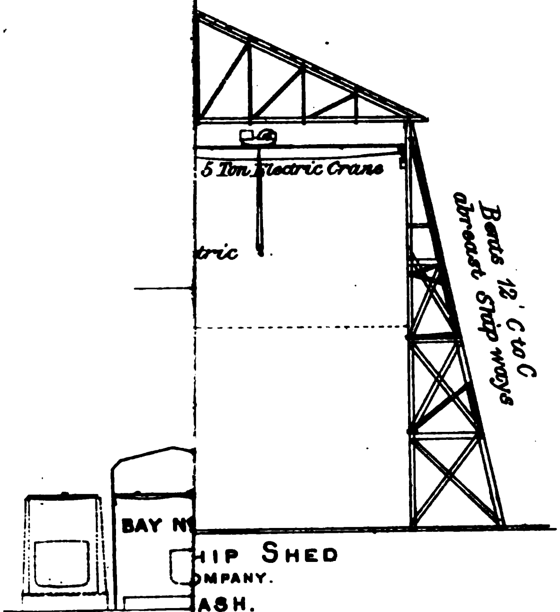
Fig. 2.

VER GANTRY CRANE. (UNION DRY DOCK CO., BUFFALO.)

ROOFED SHIPHOUSE, WITH RADIAL ARC T

Fig. 4.

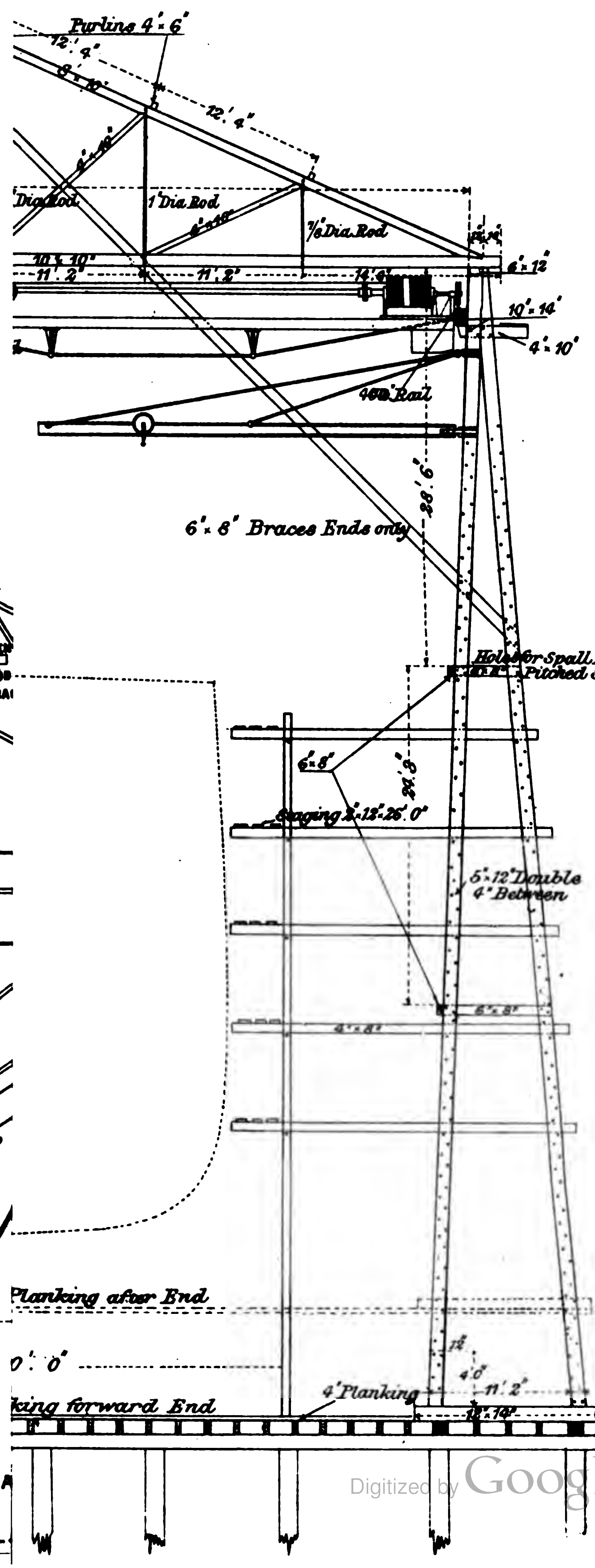
EDO BOAT "T. A. M. CRAVEN." (BATH IRON WORKS.)



TH

SCO - CAL.

FIG. 3.



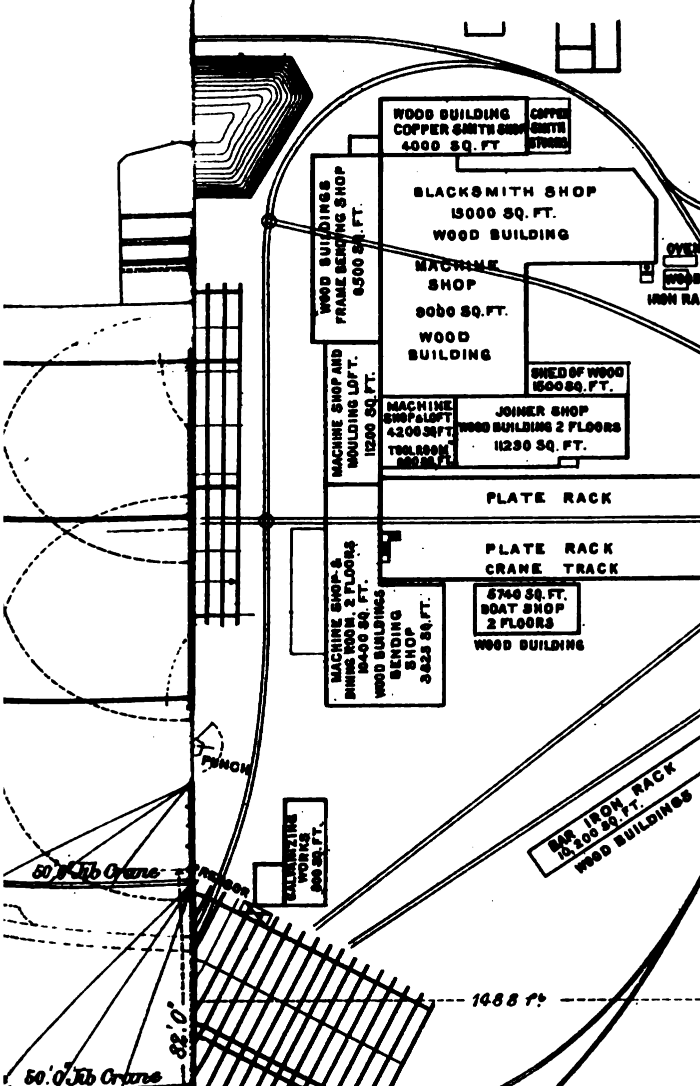


FIG. 5
ARRANGEMENT OF
SHIP BUILDING SHEDS A[…]
UNION IRON WORKS
SAN-FRANCISCO, CAL.

To Illustrate Mr. W. A. Fairburn's Paper, "Methods of Handling Material over Shipbuilding Berths in American Shipyards."

SHIPBUILDING SHEDS AT UNION IRON WORKS SAN FRANCISCO.

LE SYSTEM OVER BUILD

RN SHIPBUILDING C° NEW LO

FIG. 3

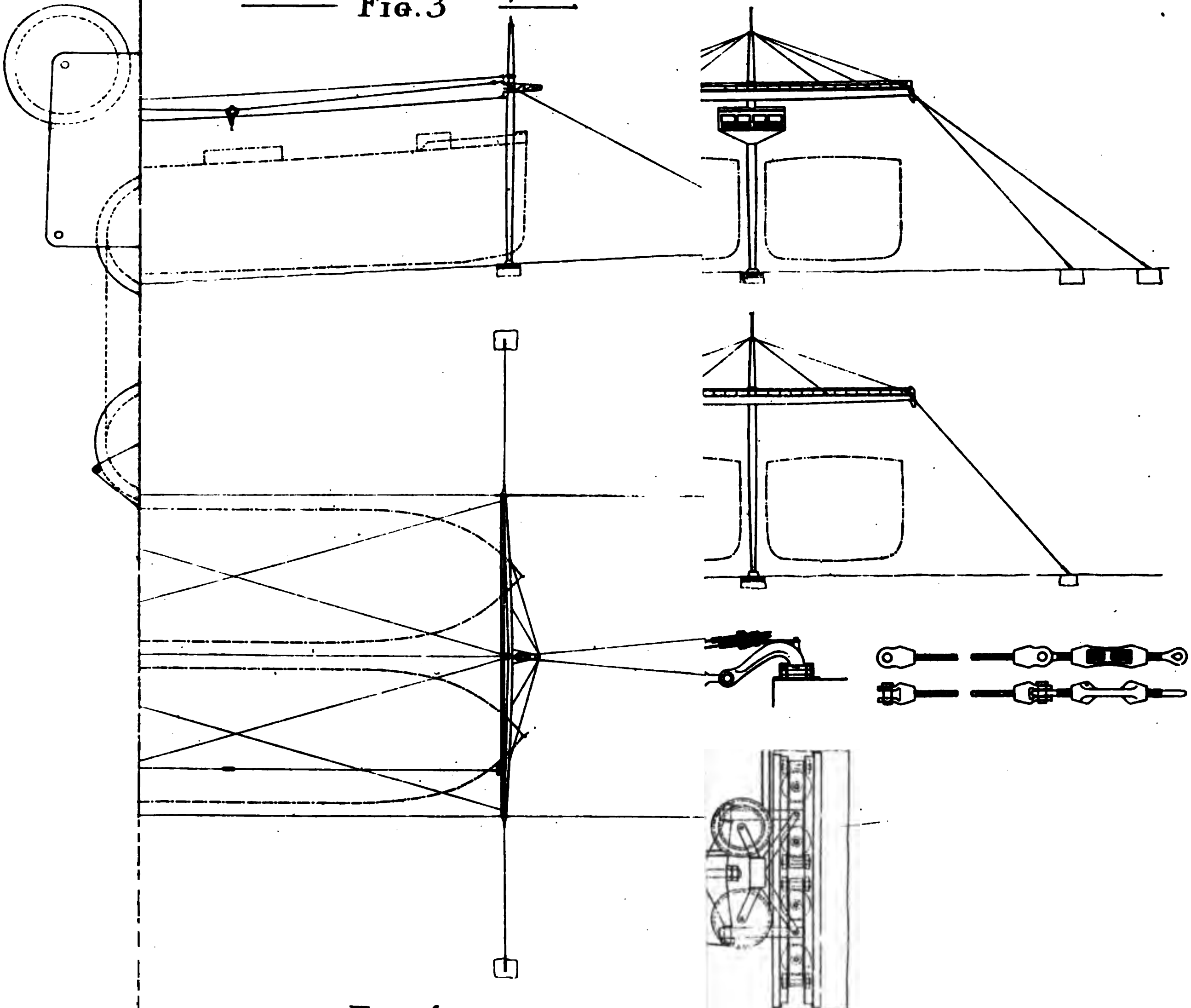
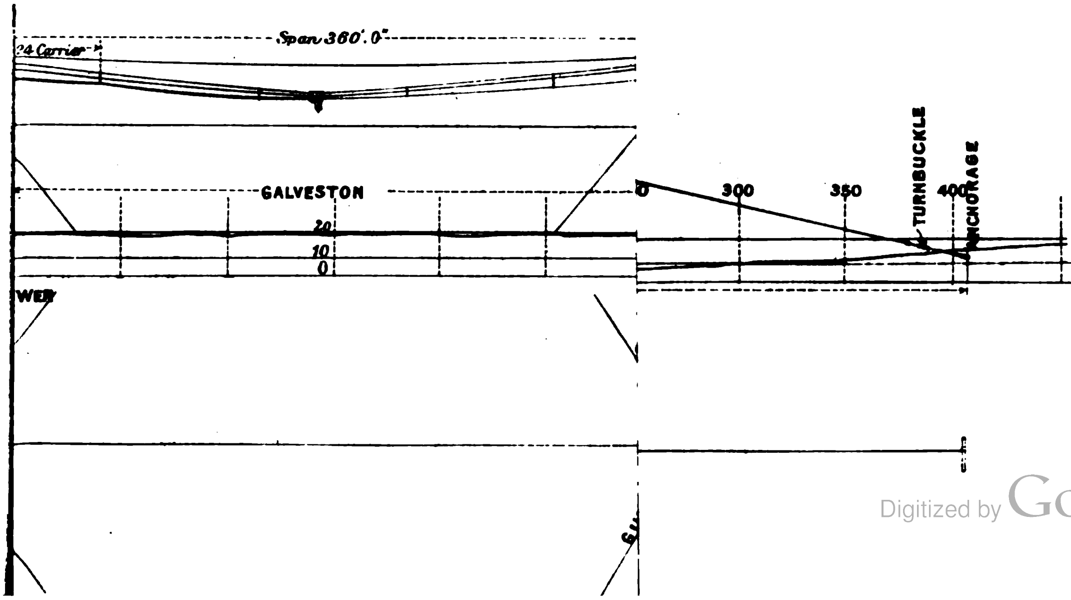


FIG. 4.

LIDGERWOOD CABLEWAY.

AT WORKS OF

W.R. TRIGG C° RICHMOND. VA.



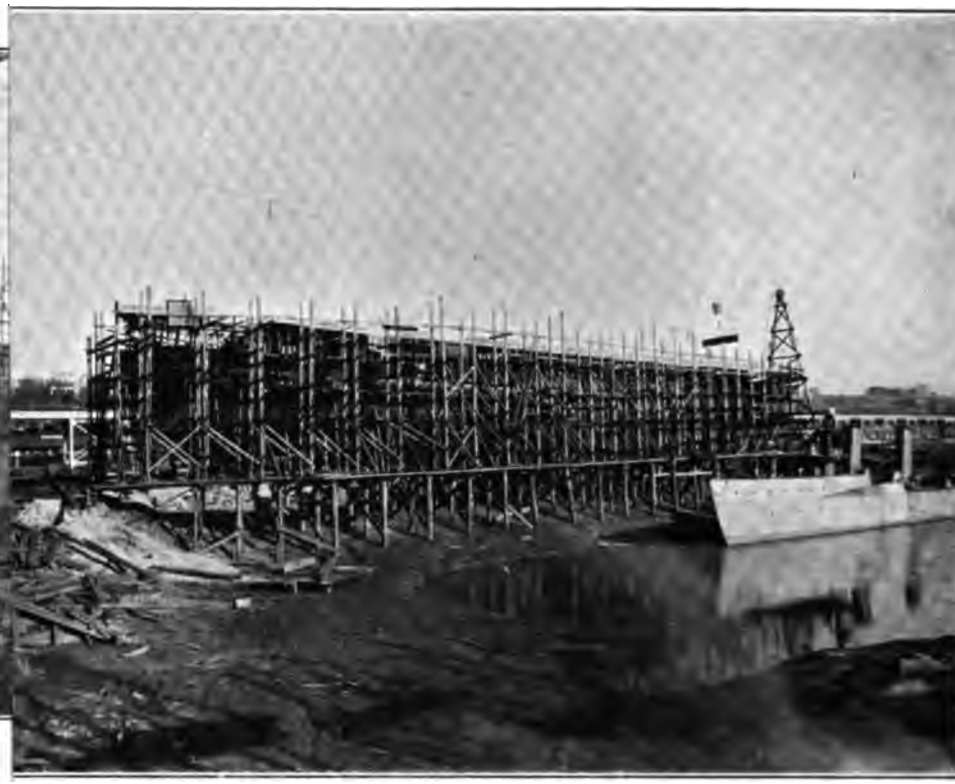


Fig. 3.
LIDGERWOOD CABLEWAY AT THE W. R. TRIGG CO., RICHMOND, VA.

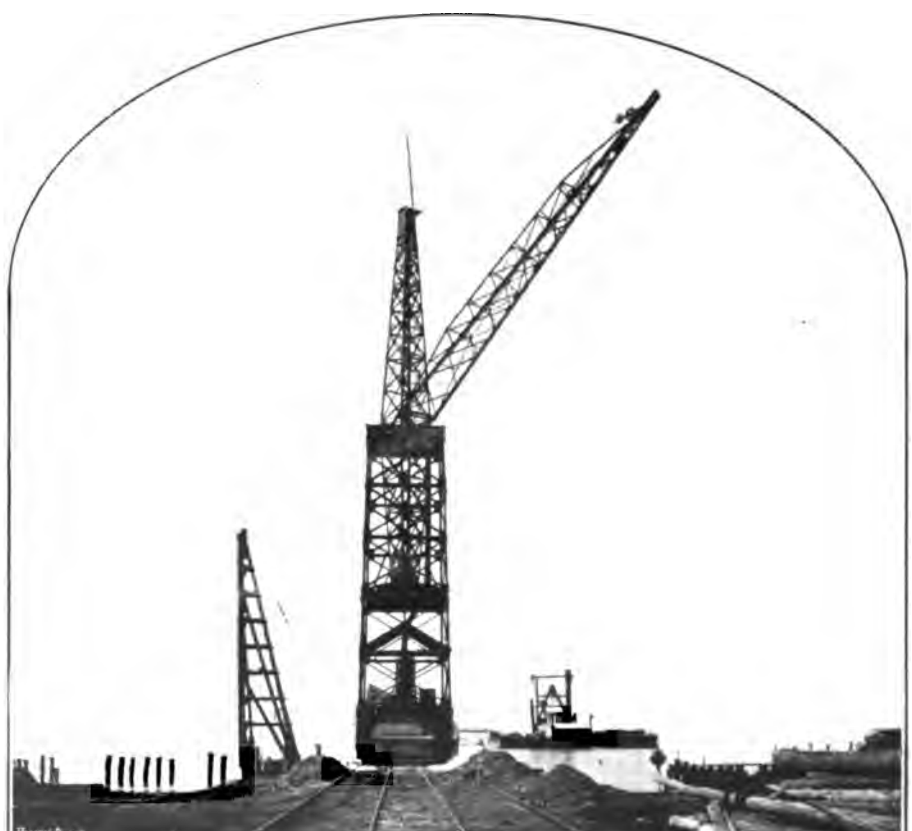
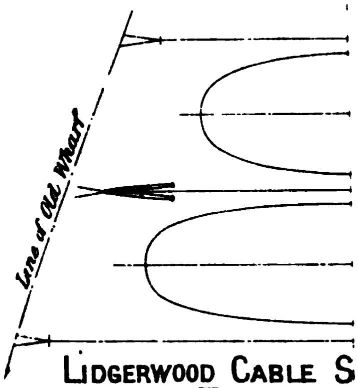


Fig. 5.

ND STEEL CO.)



LIDGERWOOD CABLE S
TOWNSEND
SHOOTERS

FIG. 1